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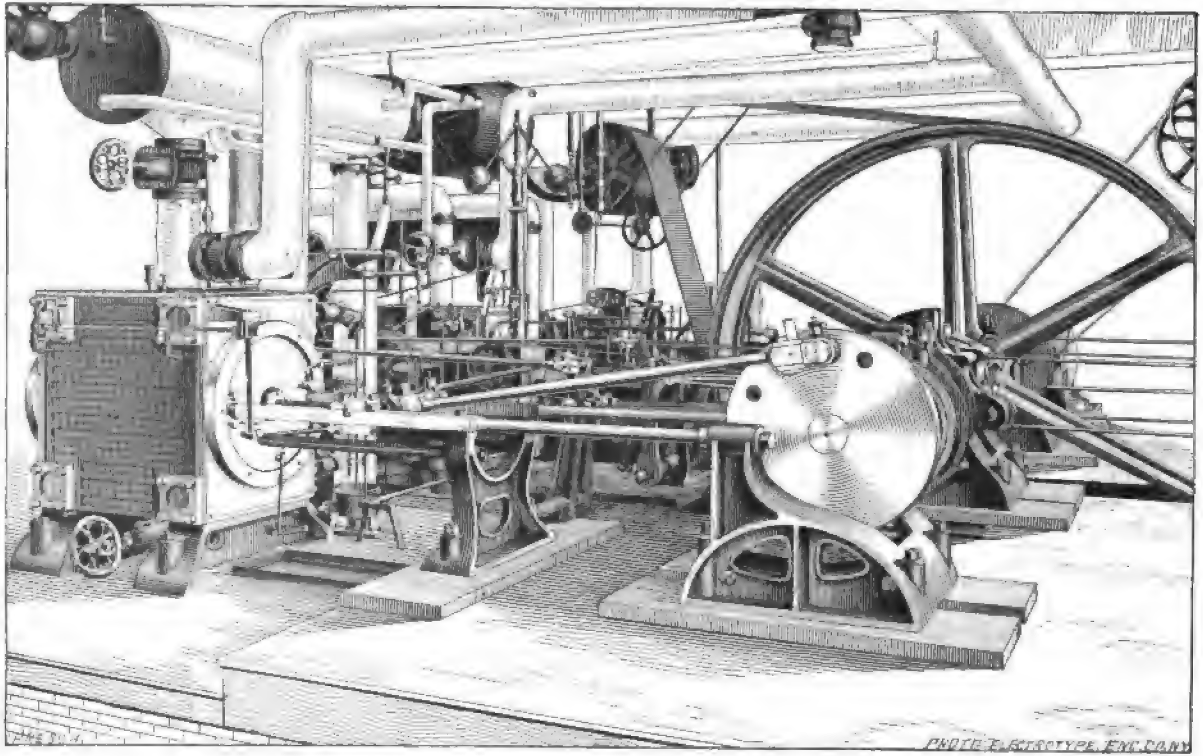
PART II.

DESIGN, CONSTRUCTION, AND OPERATION

OF THE

STEAM-ENGINE.



EXPERIMENTAL ENGINE.

Frontispiece.

A MANUAL

OF THE

STEAM-ENGINE.

PART II.

DESIGN, CONSTRUCTION, AND OPERATION.

FOR ENGINEERS AND TECHNICAL SCHOOLS

(ADVANCED COURSES).

BY

ROBERT H. THURSTON, A.M., LL.D., DR. ENG'G;

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NEW YORK:

JOHN WILEY & SONS,

53 EAST TENTH STREET.

1891.

ROBERT DRUMMOND,
Electrotyper,
444 and 446 Pearl St.,
New York.

FERRIS BROS.,
Printers,
326 Pearl Street,
New York.

PREFACE.

THE first part of this MANUAL OF THE STEAM-ENGINE, already published, constitutes, in some sense, an independent work, and may be considered an epitome of the purely scientific side of the subject, an *exposé* of the theory of the steam-engine considered as a thermodynamic machine, and its efficiencies as such. Its purpose is twofold: (1) the development of the mathematical processes which enable the engineer to trace the flow of energy into and through the engine, and to exhibit the direction of its variously distributed streams and their useful or wasteful disposition; (2) the application of the facts, principles, and equations and formulas of thermodynamics and of mathematical physics, thus collected into a system and a theory, to the computation of the quantities of heat, steam, and fuel to be required for the production of power in a specified engine, and, further, the determination of just what proportions of engine and what distribution of steam will probably give the highest commercial result. Such a treatise—in the present instance perhaps too extensive for any but the highest order of technical schools—may serve well for the purposes of instruction in the senior classes of undergraduate courses in engineering; more or less satisfactorily taking a place such as was, in earlier days, so exceptionally well filled by the older works on applied thermodynamics. For the specialist and the practitioner, however, this is but the first step in a much larger course of study and investigation. For his purposes, this introductory must be followed by a somewhat extended study of principles and current practice in the design, the construction, and the operation of the

machine. It is this division of the subject which is treated of in this second part of the work, and which is considered to form a proper supplement to the more purely scientific treatise which has preceded.

The second part of this MANUAL OF THE STEAM-ENGINE is thus intended to present, as concisely as possible, the later and more usually practised methods of design, construction, and operation of the engine, as based upon the technical knowledge and the professional experience of contemporary specialists, and as illustrated in the best current practice. The relation between the two parts, as well as their details, may be seen by reference to the "Plan," on page xv. The first part is seen to contain the scholastic foundation, the second the practitioner's superstructure. The first is a book for the study; the second for the office. Both may serve, it is hoped, in the professional education of the novice just entering the profession with a special inclination to the pursuit of this line of work in so immensely extended a field.

The Author has been led to attempt the preparation of this portion of the work in the endeavor to provide suitable instruction for graduate students in mechanical engineering—men who have completed the somewhat extended and purely professional course of late years taught in Sibley College, at Cornell University, and who have, either with or without a period of intermediate professional practice, entered upon a postgraduate course of higher character and more extensive range. Such a course as is comprehended in these two volumes, unsatisfactorily brief as it necessarily still remains to probably many practitioners, could only be offered, in the college, to graduate students, and only to such among them as had previously taken a strong course of undergraduate instruction. Four years of purely professional studies, based upon entrance requirements including the higher mathematics, and a good high-school preparation, as demanded in the case here contemplated, should place the young engineer safely at the desired point for beginning really graduate work. This is now becoming a common experience; and it is the demand so

coming to the Author that has led to this endeavor to meet it by the preparation of this work. The two parts are intended to form a full one-year's course of work in steam-engineering for graduate students.

It is by no means likely that the expert engineer will consider even this course a complete one. It should be supplemented, further, by the study of the best practice of the day, as illustrated in actual constructions, and by working drawings of the latest and best of modern machinery of this class. It is the hope of the Author—should it prove that he has been reasonably successful in his effort to devise a suitable plan, and to give it form—time permitting, that he may, later, be able to provide an atlas of such drawings, with concise accompanying text, as a third part in this series. Meantime, the admirable work of Mr. Daniel Kinnear Clark will be found to contain such a collection in illustration, especially of British practice, and the young engineer taking advantage of his opportunities of observation of current practice at home will find such object-lessons the best possible means of instruction.

The volume here offered the advanced student and the practitioner will be found to contain a condensed description of the operations involved in the design, construction, and management—including preservation and repair—of the steam-engine. Even so large a volume as this has become can be made to contain no more. The introductory chapter on design includes the general discussion of the proportions of the compound and other multiple-cylinder engines, as well as that of details. The great development of this class of machine is well shown by the variety of groupings and adjustments of the several cylinders which take up so much space in this chapter. The principles of thermodynamic action have already been exhibited in PART I. The study of the valves and their gears is made mainly by the processes of Zeuner and other graphical methods; as the engineer invariably takes graphical in preference to algebraic methods, where choice is allowed, and finds, as a rule that his work is thus greatly facilitated. By reference to the special works on the subject, as those of Spangler,

of Halsey, of Auchincloss, and others, in English, and of Blaha, Uhland, Zeuner, and others among Continental writers, any deficiencies noted by the practitioner seeking more extended treatment may be made good. In the proportioning of details, the principles of resistance of materials are followed; while the results are checked by reference to that higher instruction only obtainable by actual experience in the operation of the machine. The requirements of practice include consideration of the effects of many other conditions than those of strength of materials to resist simple and ordinary stresses. The effects of inertia, especially in case of accidental accelerations, such as have been known to destroy the great engines of an ocean-steamer, and the possibility of extraordinary contingencies, as of water entering the steam-cylinder, must be taken into account; and these can usually only be measured through observation and experience of their often serious results. It is for this reason that it often happens that formulas for such proportions are empirical, and even conventional, the general form being merely tentatively given by theoretical considerations, and their numerical quantities being obtained only after long experience of working conditions, such as determine length of life and liability to accident. The added constant, in the case of the common formulas for size of piston- and connecting-rods, illustrate this case.

In the discussions of the effects of inertia, as exhibited in the action of the fly-wheel and of the reciprocating parts, the graphical methods have been preferred, where choice was permitted, for the reasons already given. The algebraic equations give isolated values of the quantity sought, also; while the curves obtained, in many cases, by graphical construction, are a visible history of the continuous variations of condition and result throughout the cycle, or the period studied. The physical "concept" is thus vastly more easily and satisfactorily obtained, and a correspondingly better understanding of the case is reached.

The chapters on the construction and erection, and on the operation and care, of the engine are necessarily very concise;

but they are based upon long practical experience, observation, and application, and it is hoped that they will prove to constitute at least a good outline, to be filled in later by personal experience. The account of materials used is abridged from a larger work by the Author.* The account here given of the modern methods of trial of engines and boilers is abstracted, mainly, from the Author's work on that subject; and the specialist may perhaps fill in this skeleton account from that treatise.† The chapter on specifications and contracts is intended to be an exceedingly condensed discussion of that important subject. The reader will find in Haupt on Contracts a more complete treatment.

The last chapter is one which will perhaps excite more interest, and very probably more criticism, than any other in the book. It is a first attempt to introduce the financial element into the theory and practice of steam-engine construction; and probably no one is more fully aware than the Author of its incompleteness, and the defects of form and data arising from that fact. After a time, when engineers have become more generally accustomed to considering these elements of success in practice, and shall have collected more complete and more accurate statistics, it will be comparatively easy to secure a systematic and readily applicable discussion. Designing for a minimum cost is a matter which every experienced member of the profession has constantly been compelled to consider; but scientific methods of computation of minima rarely have been known or practised. Like the final chapter of PART I of this work, this concluding discussion must be regarded as still in its initial stage and likely to experience great modification, and probably as great improvement, at an early date. It is one of those matters which only requires to be formally placed before the profession to insure its careful consideration, systematic study, and complete final development. So important has this subject seemed to the Author, and from a very

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* **Materials of Engineering: Vol. II. Iron and Steel; Vol. III. The Alloys and their Constituents. N. Y., J. Wiley & Sons.**

† **Engine and Boiler Trials. N. Y., J. Wiley & Sons.**

early period in his professional career, that his first scheme for a course of professional education (May, 1871) included the study of costs at every step. As a matter of now historical interest, this scheme is given in the Appendix.* Should this work meet with the favor accorded to the preceding volumes from the pen of its Author, it will be endeavored, in later editions, to take full advantage of the opportunity to incorporate any such improvements of form or matter as may in the interval appear.

The Author has perhaps sufficiently acknowledged his indebtedness to earlier and contemporary writers in the references and foot-notes which he has introduced into the body of this work so liberally as greatly to confuse the printer and embarrass the publishers; but he feels that he should give especial credit to a few of those into whose precincts he has somewhat frequently ventured, and whose works should be as carefully collected by the young engineer as the more pretentious treatises of the most distinguished scientific men. Such works as those of Wilson on the practical construction of steam-boilers, of Forney on the locomotive, and of Wellington—a treatise on the location of railways, but crowded with useful and reliable, as well as rare, data on engines, construction, and finance—deserve the fullest credit from any collaborator, as well as do the more extensive works of Shock, of Clark, of Ledieu, of Busley, or of the "thermodynamists." The monumental and cyclopedic work of Weisbach, as well as the many special and monographic papers and minor treatises, must be relied upon, by the young engineer and the older practitioner, to supplement what must always, in cases of this kind, be a mere preliminary reconnoissance of the field, even though extended to the magnitude of the present work; and even in one of its main divisions, that of design, such standard works as those of Unwin and of Whitham must be taken for final reference.

* From "Instruction in Mechanical Engineering"; Scientific American Supplement: April 19, 1884.

Throughout, and in both volumes, no trouble or expense has been spared in the endeavor to secure full and minute illustration. The engravings have been carefully made, and, in many cases, makers have been freely drawn upon for illustrative examples of their constructions. The publishers and their printers and binders are entitled to great credit for their admirable work, as here illustrated; work which pleases the reader as well as the Author. The composition has been a marvel of accuracy on the part of the printer, and the excellence of the typography, as well as of the bookmaking itself, the printing and binding, together with the freedom of illustration permitted by the publishers, are such as to entitle them to very great credit.

The Author is also under especial obligations to those friends, and those fair and kindly critics, who have, in the most considerate manner possible, aided him by indicating faults of omission and of commission. It is hardly less satisfactory to secure the correction of a defect or of an error in the text than to find that part of the work, original with its author, has proved of valuable service to builders: as when he is informed that the proportions of journals which he had deduced are made standard by successful makers of "high-speed engines." Even an occasional thrust from a less courteous seeker of faults is not unwelcome; for frequent experience indicates that, though sometimes in the wrong, the critic may often enable the author to improve his work greatly, or to find new ways of making his labors useful. Such criticism is often both useful and wholesome. The Author of this work will always be grateful to his friends for their kind assistance in this direction, and it is hoped that, should succeeding editions be called for, many ways may be found of securing improvement of plan and of material.

DIRECTOR'S ROOMS, SIBLEY COLLEGE, CORNELL UNIVERSITY,
ITHACA, N. Y., December 15th, 1891.

A MANUAL OF THE STEAM-ENGINE.

PLAN.

PART I. STRUCTURE AND THEORY.

PART II. DESIGN, CONSTRUCTION, OPERATION.

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A MANUAL OF THE STEAM-ENGINE.

PART II.

CHAPTER I.

DESIGN OF THE STEAM-ENGINE.

1. **The General Design of the Steam-engine** involves the study of the prescribed conditions under which it is to be operated. Such are, for example, its location and the amount and character of its work; the costs of the engine and its accessories, and the expenses incidental to its operation; the steam-pressure proposed or permissible; the relative value for the place and purpose of low- and high-speed, of vertical or horizontal engines, of simple or complex gearing; and the nicety of regulation demanded. These facts and data are usually easily ascertainable by the designing engineer, if not supplied by his employer.

The considerations determining the type to be adopted in any given case, in which the engineer is called upon to design an engine for a specified locality and purpose, are thus such as relate to space occupied, weight of engine and often of fuel, and all the economical conditions which have been studied as part of the theory of actual engines. It is frequently the fact that the conditions affecting the case are such as to produce a very definite type, and one which, while peculiarly well fitted to the

specified place and purpose, could not well be substituted for a type equally definitely adapted to another application. It is thus that the locomotive and the marine engine have always been made to differ, and the one is never made to do the duty of the other; nor is a factory engine ever removed from its proper location to do duty in place of either. The stationary; the locomotive; the marine engine: each forms a distinct and well-recognized type; and, as a rule, when the duty to be performed is prescribed within this classification, the general character of the machine is at once settled.

Similarly, within each class, circumstances and conditions of operation determine variations of type which are often as well recognized as the classes to which they belong. For example, among stationary engines, the common and the "high-speed" engines; the "positive-motion" and the detent or "detachable" valve-motions; the "automatic," the "throttling," and other engines: all are distinctly defined genera of the general type. Each is best suited to some special environment and set of economical conditions. Among locomotives, the most common and generally useful form is the ordinary mixed passenger and "freight" or "goods" engine, with its two pairs of coupled "drivers," its forward four-wheeled "truck" or "bogie," supporting the smoke-box end of the boiler, its "cab" protecting the engine-driver and the fireman or "stoker," and its attached "tender," carrying its fuel and water supplies. As the loads become heavier, however, the number of driving-wheels is increased; their diameter is diminished; the "truck" is eliminated, first by the substitution of a simple pair of leading wheels, then these are given up, and the machine becomes the enormously heavy and very differently appearing engine used purely for heavy gradients and traffic, at moderate speeds. Change of conditions in the opposite direction produces the large "express," or "fast" engine, with its larger wheels and special proportions for great speed; while light loads and short runs are productive of the small tank-engine which carries its own supplies of fuel and water and has no "tender." From among marine engines, the merchant service ordinarily calls for

the vertical direct-acting engine, the naval service for horizontal engines capable of being stowed below the water-line. Coasting steamers are fitted with comparatively inexpensive engines; and transoceanic lines demand efficiency, economy in the use of fuel, compactness and reliability, without regard to cost, and work their engines day after day unceasingly by the week together; while the torpedo-boat requires enormous power in proportion to space occupied and weight carried, but only for a few hours at a time.

Other things equal, an engine is valued in proportion as it exhibits power combined with lightness. The chief advantages are usually reckoned as (1) ample power; (2) minimum cost of operation; (3) small expenses in repairs; (4) minimum interest, tax, and insurance accounts.

With some kinds of work, as in fine spinning and in electric lighting, good regulation is only second in importance to ample power.

The Conditions of Success with either type or form are, in general, identical. As regards the construction of the engine as a piece of mechanism, the principles and practice of good engineering are precisely the same, whether applied in the designing of one or another type of engine. Conditions of success in practice may be stated to be, for all forms of engines, as follow:

(1) A good design, by which is meant—

a. Correct proportions, both in general dimensions and in arrangement of parts, and proper forms and sizes of details to withstand safely the forces which may be expected to come upon them.

b. A general plan which embodies the recognized practice of good engineering.

c. Adaptation to the specific work which it is intended to perform, in size and in efficiency. It sometimes happens that good practice dictates the use of a comparatively uneconomical design.

(2) Good construction, by which is meant—

a. The use of good material.

b. Accurate workmanship.

c. Skilful fitting and a proper "assemblage" of parts.

(3) Proper connection with the work to be done.

(4) Skilful and careful management.

It is evident that this adaptation of the steam-engine to great speed of piston is work to be done by the engineer. The requisites to success are obvious, and may be concisely stated as follows:

(1) Extreme accuracy in proportions.

(2) Perfect accuracy in fitting parts to each other.

(3) Absolute symmetry of journals.

(4) Ample area and maximum durability of rubbing surfaces.

(5) Perfect certainty of an ample and continuous lubrication.

(6) A nicely calculated and adjusted balance of reciprocating parts.

(7) Security against injury by shock, whether due to the presence of water in the cylinder or to looseness of running parts.

(8) A "positive-motion" cut-off gear.

(9) A powerful but sensitive and accurately-working governor, determining the degree of expansion.

(10) Well-balanced valves and an easy-working gear.

(11) Volume of "dead-space," or "clearance," properly adjusted to suitable "compression."

In general, to secure maximum economical efficiency, steam should be worked at as high a pressure as practicable, and the expansion should be fixed as nearly as possible at the point of maximum ultimate economy for that pressure. With considerable expansion, steam-jacketing or moderate superheating should be adopted, to prevent excessive losses by internal condensation and re-evaporation, and expansion should take place in double cylinders, to avoid excessive heat-wastes, great weight of parts, irregularity of motion, and loss by friction. External surfaces should be carefully covered by non-conductors and non-radiators, to prevent losses by conduction and

radiation of heat. It is especially necessary to reduce back-pressure and in condensing engines to obtain the most perfect vacuum possible without too seriously reducing the temperature of hot-well and overloading the air-pump, if it is desired to obtain the maximum efficiency by expansion; it then becomes also necessary to reduce losses by "dead-spaces" and by badly-adjusted valves. The piston-speed should be as great as can be sustained with safety. The expansion-valve gear should be simple. The point of cut-off should be determined by the governor. The valve should close rapidly, but without shock, and should be balanced, or other device should be adopted to make it easy to move and free from liability to cutting or rapid wear. The governor should act promptly and powerfully, and should be free from liability to oscillate, and to thus introduce irregularities which are sometimes not less serious than those which the instrument is intended to prevent. Friction should be reduced as much as possible, and careful provision should be made to economize lubricants as well as fuel.

Guided by these general and fundamental principles of steam-engine practice, it is perfectly practicable to design engines which may be relied upon to give good economical results.

In computing sizes of parts controlled by the principles of strength of materials simply, any good treatise on that subject may be taken as a guide; * but, as will be presently seen, it is often difficult to determine precisely the nature and magnitude of the forces acting on a given piece, and some parts must be designed on quite other principles.

A process of experimental variation of type is continually going on among users of engines which leads to the selection of the best for the special case or kind of work and the rejection of less well-adapted designs. This process it is which has given form to standard locomotives, and the better classes of stationary engines, and has led to the adoption in the merchant marine

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* See "Resistance of Materials" by Burr, or Wood's treatise; also the Author's "Materials of Engineering," vols. II and III, his "Manual of Steam-boilers," chap. II; or "Text-book of Materials," chaps. XII–XVII.

of the vertical, rather than the horizontal or inclined, engine. Their more convenient stowage, the longer stroke attainable, their consequent lesser friction and clearance, and their accessibility and other minor advantages have, in the latter case, given the vertical engines the preference.

These preliminaries being settled, we may take up the question—

What is the problem, stated precisely and in its most general form, that engineers have been attempting here to solve?

After stating the problem, we are to endeavor to determine what direction the path of improvement has taken hitherto; and, so far as we may judge the future by the past, by inference, to ascertain what appears likely to be its course in the present and in the immediate future. Still further, we may inquire what are the conditions, physical and intellectual, which best aid our progress in perfecting the steam-engine. This important problem may be stated in its most general form thus:

To construct a machine which shall, in the most perfect manner possible, convert the kinetic energy of heat-motion, as derived from the combustion of fuel, into mechanical power, using steam as the receiver and conveyer of that heat.

The problem embodies two distinct and equally important inquiries: The first, What are the scientific principles involved in the problem, as stated? The second, How shall we construct a machine that shall most efficiently embody and accord with not only known scientific principles, but also with all well-settled principles of engineering practice?

In the construction of modern engines, the steam-cylinder is always, necessarily, made of cast-iron, as is the piston; no other metal, economically available, being practically suited to these parts. The same is true of the engine-frame, as a rule; although it is sometimes partly constructed in forged metal. The piston-rings are usually of cast-iron, but occasionally, especially if the cylinder is soft, of iron or brass, lined with white metal. The rods are of iron or of soft steel, the latter being more and more common. In special cases, as in steam-hammers, hard steel is employed, containing carbon to the ex-

tent of 0.6 to 0.8 per cent. The cross-head is often cast-steel, more frequently cast-iron, depending, in part, on shape. The pins of cross-head, crank, and minor connections are now often of soft steel, and sometimes case-hardened to insure good wearing qualities combined with resistance. The connecting-rod is usually of iron, but often of soft steel. Its "brasses" are usually "babbitted." The crank and shaft are often forged together of either iron or soft steel, or are a steel casting. If detached, the crank is a cast-iron disk in fast engines, or forged in other types.

The materials ordinarily employed in the steam-engine are:

(*a*) Crucible or cast or tool steel.

(*b*) Bessemer and open-hearth steel. These are usually called machinery or soft steel.

(*c*) Malleable or wrought-iron.

(*d*) Cast-iron.

(*e*) Malleablized cast-iron.

(*f*) Steel castings.

(*g*) Brass or bronze.

(*h*) Babbitt-metal. This name is used to cover all grades of white metal used for lining bearings.

These have, generally, the following qualities:

Material.	Tensile Strength.	Comparative Strength.	Resilience or Shock-resisting Power.	Shaped for use by
(*a*)	Very high	Very high	Medium	Forging
(*b*)	High	High	High	Forging
(*c*)	Medium	Medium	High	Forging
(*d*)	Low	Very high	Low	Casting
(*e*)	Medium	.	High	Casting
(*f*)	High	High	High	Casting
(*g*)	Low		Medium	Cast'g or for'g
(*h*)	Very low			Casting

The following is a tabular statement of the working qualities of the metals employed in engine-construction, prepared by Professor A. W. Smith:

PROPERTIES OF THE METALS.

	Per cent Carbon.	Stress at Elas. Lim. Tens'n.	Stress, Ultimate. Tension.	Stress at Elas. Lim. Comp'n.	Stress, Ultimate. Shearing.	Stress in Outer Fibre. Flexure.	Modulus of Elasticity. Shearing.	Modulus of Elasticity. Tension.
Bessemer and Siemens-Martin steels.	0.15	42,000	63,000	39,000	48,000		9,000,000	30,000,000
	0.20	47,000	68,000	43,000	53,000			
	0.50	48,000	80,000	46,000	57,000			
	0.70	53,000	89,000	53,000	60,000			
	0.80	57,000	103,000	63,000	68,000			
	0.96	69,000	118,000	71,000	83,000			
High-grade wrought-iron.		28,000	56,000	28,000	40,000	Elas. lim. 52,000	9,000,000	28,000,000
Common wrought-iron...		22,000	40,000	22,000	32,000		9,000,000	28,000,000
Crucible or tool steel. ...		58,000	116,000	58,000			14,000,000	40,000,000
Malleable cast-iron.......			35,000					
Steel castings.............		29,000	47,000	29,000		Ultimate.		19 to 31 mil. Average 25,000,000
Cast-iron...............		8,000 to 10,000	10,000 to 35,000 20,000 av.		18,000 to 20,000	30,000 to 54,000 42,000 av.	7,000,000	13,000,000

2. **The Selection of the Type** of engine to be designed and constructed is thus the first step to be taken. Where the controlling circumstances dictate high efficiency, a comparison of the costs and performance of existing engines will determine roughly their relative standing for the purpose in view. The construction and comparison of their "curves of efficiency," * if data are at hand or available for them, will enable the designing engineer to solve the several problems of efficiency and thus to secure approximate accuracy of comparison. The closer the actual curve of efficiency approximates to the curve of ideal mean pressures, the higher the economical value of the type; and a set of real curves of efficiency being laid down, the relative standing of the several engines of which they are representative may be easily determined.

For general purposes, engines of moderately high piston-speed, with detachable cut-off gear; with separated steam and exhaust valves at the ends of the cylinder; and with regulation effected by a governor so attached as to momentarily adjust

* The Curve of Efficiency is one of which the coördinates measure the work done and its costs, as is seen in another chapter.

the point of cut-off and ratio of expansion, are likely to prove most suitable. As costs vary, the type and construction are changed to approximate more closely to the ideal, or in the direction of simplicity and smaller cost, as the case may demand. The most efficient thermodynamic machine is usually a jacketed multiple-cylinder engine, in other respects of the class just described; since it gives, at the same time, high economy and close regulation. With high speed of rotation, and nice regulation, as prime requisites, the later type of "automatic" engine is adopted. Cheapness being the salient qualification, the older forms of single-valve engine are chosen.

When large powers are called for, it is usually considered wise to adopt two engines coupled with cranks at right angles, or even three with cranks at 120°; unless the speed of revolution is such as to permit, in the first case, setting of the cranks opposite to secure dynamic balance and a convenient system of compounding.

The principles that must govern the engineer, in the attempt to select the most efficient type, have now been seen to be, in short:

(1) The greatest practicable range of commercially economical expansive working of steam. The fluid must enter the cylinder at the highest admissible pressure, and must be expanded down to the minimum economical pressure at exhaust.

(2) The wastes of heat must be made a minimum. All loss of heat by conduction and radiation from the engine must be prevented, if possible, and the usually much more serious waste which occurs within the engine, by transfer of heat from the steam side to the exhaust, by "cylinder-condensation" and re-evaporation, without doing its proportion of work, must be checked as completely as is practicable. This latter condition, as well as commercial considerations, limits the degree of expansion allowable. It also dictates high speed of engine.

(3) The largest amount of work must be done by the engine that it can perform, with due regard to the preceding conditions. This condition compels us to drive the engine up to

the highest safe speed, and to adopt the highest practicable mean steam-pressure.

The first two of the above requirements give maximum efficiency of fluid, consistent with commercial economy, and the latter gives highest efficiency of machine. In addition to these requisites, which are not peculiar to any style of engine, or to any one of the innumerable applications of steam-power, the adaptation of the machine to its work often, as, for example, in driving dynamo-electric apparatus, compels the engineer to meet certain demands which, although not peculiar, are, nevertheless, more imperative than elsewhere. The principal of these requirements are effective regulation, united with compactness, simplicity, strength and durability, and small cost, both of original purchase and of repairs.

In designing the "drop cut-off," or "automatic" engine, care should be taken to secure the proper size of engine, either by the methods described in the chapter on engine efficiencies, or by reference to the results of experience with the type of engine chosen. In detail, the steam should be employed as nearly as practicable at boiler-pressure, and the steam-pipe and valve-motion should be so proportioned that the piston may be actuated, quite up to the point of cut-off, by as nearly boiler-pressure as possible. Loss of pressure due to small pipes, contracted and tortuous passages, and ill-designed valve-gear, is productive of very noticeable loss of resultant economy.

Clearance should be made a minimum, and the dead-spaces should be filled by compression at the end of the return-stroke; while wastes due to internal condensation should be made a minimum by proportioning the engine for high speed of piston and, where practicable, by jacketing, superheating or compounding, or by combining these expedients. The exhaust-steam should be led away through passages of which the walls should be separated from the cylinder-casting in such manner that the latter may not be chilled by them. Ample size of exhaust passages and pipes, and especially through the heater, if one be used, should be provided in order to make the back-pressure the least possible.

Steam and exhaust valves should open quickly and uncover an ample port-area, and should close with even greater rapidity; so that no serious "wire-drawing" or "throttling" of the steam may take place; and so as to permit the steam line on the indicator-diagram to be as nearly as possible horizontal from end to end, with a clean sharp corner at the point of cut-off, and a back-pressure line equally smooth, straight, and horizontal.

Where independent steam and exhaust valves are used, it is advisable to make them separately adjustable with ease and accuracy. The friction of the valves and gear should be reduced as far as possible and the movement of the cut-off gear should not throw a sensible and irregular load on the governor; which should be sensitive, yet stable and powerful. Such parts as are subjected to heavy loading should be made so stiff as not to yield and thus, by springing, throw greater loads on already strained parts, or increase friction.

Uniformity of rotation during each revolution is secured by providing a fly-wheel of ample size.

Heavy frames and attached parts, and light running parts, are usually best, and cheapest in the end. The free use of steel in the latter is a decided advantage. At least 30 per cent must usually be added to the demanded power to allow for intermediate losses at the countershafts, belting, and other machinery of transmission.

All parts should be as simple in form and as inexpensive in construction as is consistent with efficient operation; all should be accessible for inspection, lubrication, and repair; and those subject to wear should be so arranged as to be easily adjustable in order to take it up. Large wearing surfaces and their certain and free lubrication are essential.

Condensing and Non-condensing Engines are commonly selected either because of the presence or absence of sufficient available condensing water, or after determining their difference in cost and relative efficiencies. The addition of the condenser is almost always a means of increased efficiency and reduced expense of steam and fuel; but it is often, especially in

cities, either impracticable to secure a good and reliable supply of condensing water, or impossible to obtain it at a cost less than the saving anticipated by its use. The advantage of condensation is less, also, as pressures increase; and, with simple engines, it is rarely considered best to go to the expense of adding condensers and air-pumps, and of water supply and discharge, when the pressures are as high as those coming to be customarily employed in mills and factories. The almost universal employment of the feed-water heater, also, which is not used in the operation of the condensing engine, carries the balance on the economy account decidedly toward the side of the non-condensing engine. Where the condenser is used, the vacuum should be the less, and the temperature of the condenser and hot-well higher, as pressures are increased. At high pressures, it is probably often, if not usually, advisable to work the condenser at atmospheric pressure and dispense with the air-pump.

A surface-condenser is necessary at sea to avoid the introduction of salt into the boilers. Small stationary engines are invariably non-condensing; many large engines, and nearly all of the marine type, are fitted with condensers. Although the use of the condenser is considered impracticable on locomotives, air surface-condensers have been sometimes experimentally applied.

In the simple unjacketed engine, the gain by the condenser will often be found to be offset by the reduced cost of construction and maintenance of the non-condensing engine, especially when a good heater can be used with the latter, and thus the feed-water sent to the boiler at nearly 212° F. (0° C.) This, under usual conditions, will be very apt to prove the fact if the steam-pressure much exceed 75 or 80 pounds per square inch. With the better class of compound engines, the limit rises to some unknown but considerably higher point.

A difficulty often experienced with high rotative speeds for condensing engines is in the working of the air-pumps, when directly connected to the reciprocating parts of the main engine. It is difficult to secure an efficient air-pump making

more than 160 strokes per minute. With the pumps disconnected from the main engines, and operated independently, 5-feet-stroke engines would make 90 revolutions without any trouble, but 75 revolutions is found to be safer for continuous running with pumps attached.

Factors of Safety, and "Breaking-pieces" as they are called, in so important a class of machines, should be so proportioned and so located that a general breaking up of the whole, as by water in the steam-cylinder, for example, may not be likely to occur. The factors of safety of parts exposed to shock are usually made 8 and sometimes greater—those of parts in which solidity and stability are essential, or those made of brittle material, may be twice or three times as great, or even more,—and the magnitude of this factor should be graded from, sometimes, less than 2 in the breaking-piece to these higher figures, as the importance of the member, or the danger of and from breakage, is the greater, either intrinsically or as related to other parts of the machine. In the steam-engine, the breaking-piece is commonly a false head to the cylinder, a separate diaphragm forming part of the cylinder-head proper, or of some convenient portion of the cylinder. It is occasionally represented by a set of cylinder-head bolts, designed with a low factor of safety. It should not be either of the elements of the "running parts," as a fracture there would be very certain to lead to the destruction of other parts, if not of the whole engine.*

Where a breaking-piece is introduced, it is vitally important that it shall be so designed and located that it shall certainly perform its office. Maximum economy of construction is, in such case, secured by simply proportioning all parts of the structure so that, at the limit of resistance of the breaking-piece, they shall not be sensibly distorted. In other words, as a general rule, their "primitive" elastic limits should correspond closely with the point of rupture of the breaking-piece.

* An exception to this rule may be admitted in the case of the piston-rod, it being proposed to make this a breaking-piece by reducing it at the face of the piston.

3. **The Steam-pressure** to be adopted is determined, generally, by considerations both of safety and of economy. Economy dictates high pressures and that pressures should be higher, and the ratio of expansion greater, as the type and construction of the engine is the more perfect; while risks are necessarily increased, other things equal, as higher pressures are carried. There is a limit, economically, however, to increasing steam-pressures; and this limit is probably not far from five or six atmospheres (75 or 90 pounds per square inch) for the older engines; but it has risen to, probably, twice or three times that figure, at least, with the best modern multiple-cylinder engines. We find the lower limit adopted in common stationary-engine practice very generally; the upper limit is reached in the later ocean steamships with triple or quadruple expansion; and in the more radical work of the former class; while locomotive engines, and the standard double-expansion compound marine engine, illustrate the use of intermediate pressures. Further advance in this direction is largely dependent upon the general adoption of boilers especially designed to sustain, permanently and safely, still higher steam-pressures.

In any given case, the engineer first considers how high a pressure he may safely adopt; he then determines whether, all things considered, it is wise to construct his design on the assumption that such pressure can always be depended upon; and he finally settles the question whether an engine so designed would be a desirable one, on the score of ultimate economy, including in operating expenses the interest account and costs of maintenance and repair. He ultimately decides upon such a steam-pressure as will, in his judgment, prove, in the course of the life of the engine, safe, reliable, and of maximum resultant economy.

The ratio of expansion to be adopted in designing the engine is properly settled by the study of all the economics of the case. The best ratio, all things considered, is less than that which gives maximum efficiency of engine; and this latter is often but a fraction of that giving maximum thermodynamic

efficiency. The correct ratio to adopt is commonly that which gives maximum "commercial efficiency" in the sense in which that term is elsewhere used in this work. In the cases of ordinary steam-engines, examined by the Author when studying the modifying conditions observed in practice, the best ratios of expansion for maximum *duty*, r_e', for several well-known and typical classes of engines were taken by him as below, it being assumed that the engines are of good type and construction, and of fair size: *

PROBABLE TERMINAL PRESSURES AND RATIOS OF EXPANSION FOR MAXIMUM EFFICIENCY.

Initial Pressures. *Absolute.*	Speed of Piston.	Single Cylinders.								Multiple-cylinder; Condensing.					
		Non-condensing.				Condensing.				Sides jacketed.		Heads and sides jacketed.		Heads and sides jacketed with efficient superheating.	
		Unjacketed.		Jacketed.		Unjacketed.		Jacketed.							
		p_t	r_e	p_t	r_e	p_t	r_e	p_t	r_e	p_t	r_e	p_t	r_e	p_t	r_e
P_1	V	Lbs. on sq. in.		Lbs. on sq. in.		Lbs. on sq. in.		Lbs. on sq. in.		Lbs. on sq. in.		Lbs. on sq. in.		Lbs. on sq. in.	
40	400	20	2.0	20	2.0	16	2.5	13	3.0	9	4.5	7	6	7½	5
	625	20	2.0	20	2.0	13	3.0	13	3.0	9	4.5	7	6	7½	5
60	400	20	3.0	20	3.0	20	3.0	15	4.0	11	5.5	8	7.5	7½	8
	625	20	3 0	20	3.0	15	4.0	15	4.0	11	5.5	8	7.5	7½	8
80	400	23	3.5	20	4.0	23	3.5	18	4.5	13	6.5	9	9	7½	11
	625	20	4.0	20	4.0	18	4.5	18	4.5	13	6.5	9	9	7½	11
100	400	25	4.0	20	5.0	25	4	20	5.0	14	7.0	10	10	7½	13
	625	20	5.0	20	5.0	20	5	20	5.0	14	7.0	10	10	7½	13
120	400	27	4.5	22	5.5	27	4 5	22	5.5	16	7.5	11	10.5	7½	17
	625	22	5.5	22	5.5	22	5.5	22	5.5	16	7.5	11	10.5	7½	17
150	400	30	5.0	25	6	30	5.0	25	6.0	18	8.5	12.5	12	7½	20
	625	25	6.0	25	6	25	6.0	25	6.0	18	8.5	12.5	12	7½	20
200	400	36	5.5	29	7	36	5.5	29	7.0	20	10.0	14	14	7½	27
	625	29	7.0	29	7	29	7.0	29	7.0	20	10.0	14	14	7½	27

The terminal pressure, p_t, is taken as equal to the sum of back-pressure and friction in the non-condensing engine.

Deduct 14.7 lbs. per sq. in. = 1 kg. per sq. cm. to obtain gauge pressures. Hyperbolic expansion is assumed.

* On the Behavior of Steam in the Steam-engine, etc. Jour. Frank. Inst., Feb. 1861.

The absolute values of the weights of steam used in engines under the conditions above considered cannot be predicted with any great accuracy. A set of empirically obtained values are given in the succeeding tables, as representing good practice, and as illustrating the best work of recent builders.

The values taken accord moderately well with the observation and experience of the Author where large engines of good design have been compared, and may possibly prove useful in designing or in drawing up specifications. They can only be taken as probable means, and adopted provisionally, until better and more accurate values are determined. Thus, we have the following probable values of weight of steam demanded where the ratio of expansion is correctly adjusted for highest efficiency:

PROBABLE MINIMA.

PROBABLE MINIMUM WEIGHTS OF STEAM PER HOUR PER HORSE-POWER.

r_e	W Pounds.	W_m Kilos.	r_e	W Pounds.	W_m Kilos.	r_e	W Pounds.	W_m Kilos.
3	32	15	8	20	9	13	17	8
4	27	12	9	19	9	14	16	7
5	25	11	10	19	9	16	16	7
6	22	11	11	18	9	20	15	7
7	20	9	12	17	8	25	16	7

PROBABLE MINIMUM WEIGHTS OF COAL PER HORSE-POWER PER HOUR.

r_e	W' Pounds.	W_m' Kilos.	r_e	W' Pounds.	W_m' Kilos.	r_e	W' Pounds.	W_m' Kilos.
3	3.5	1.6	8	2.2	1.0	13	1.9	0.9
4	3.0	1.4	9	2.1	1.0	14	1.8	0.8
5	2.8	2.3	10	2.1	1.0	16	1.8	0.8
6	2.3	1.1	11	2.0	0.9	20	1.7	0.8
7	2.2	1.0	12	1.9	0.9	25	1.7	0.8

In the above, the probable minimum expenditure of coal per hour and per horse-power is taken at *one ninth* the weight of steam demanded, condensation being assumed.

For cases in which the boiler gives an evaporation of ten pounds of water per pound of coal we may look for ten per cent better figures.

Comparing these quantities with the premised gain by expansion for the ideal engine, and with the figures for weight of steam in the ideal case, the wide discrepancy between the ideal and the real cases is very impressively exhibited.

In the case of well-designed and skilfully managed condensing engines, compounded and jacketed, it has been found that those giving highest duty have usually expanded steam down to a terminal absolute pressure of from one half to two thirds of one atmosphere (7.5 to 10 lbs. per sq. inch).

The actually best ratios, from the proprietor's point of view, are lower than the above, as is seen in the table given in the chapter on efficiencies of engine.

The problem should, however, be always solved for each case taken up, if practicable.

The Speed of Engine, both of piston and of rotation, as has been seen, is continually, although not steadily, increasing. The choice of speed to be adopted in an engine about to be designed is one of the first conditions to be settled, as it is an essential element of subsequent calculations. The pressure of steam and the ratio of expansion having been determined upon, the mean effective pressure, per unit of area of piston, is fixed, and the total quantity of work demanded in the unit of time is divided by this quantity to obtain the volume to be traversed by the piston in that period. The speed of piston is a factor of that volume, which being known, the area of piston required becomes known. It next remains to determine either the length of stroke or the number of revolutions of the crank to be made in the unit of time, and this fixes the size of steam-cylinder.

James Watt's rule for V, the speed of piston, was, in terms of stroke, L,

$$V = 128 \sqrt[3]{L},$$

in feet per minute when the stroke was in feet, or 64 revolutions per minute for one foot stroke, and 256 feet per minute, or 32 revolutions, for 8 feet stroke of piston. Modern engines often exceed four times this quantity, and

$$V = 500 \sqrt[3]{L}$$

would represent a fairly high speed. The higher this velocity, the smaller and cheaper the engine for a given power, and, other things equal, the more economical. Risk of injury to bearings and danger of "breaking down" increase rapidly with increasing speeds; and it is only when excellent design, the best of workmanship, and thoroughly skilful management are combined that what are now called high speeds can be adopted with any degree of prudence.

For any assumed case, the power being given, we have

$$LN = V = a \sqrt[3]{L}, \quad \text{and} \quad N = \frac{a}{L^{\frac{2}{3}}}.$$

The speed of piston has been found to vary, in practice, nearly as the cube-root of the length of stroke; hence, the speed of rotation varies, in such cases, inversely as the two-thirds power of the stroke, and a common speed, in revolutions per minute, N, is, with a "positive" valve-gear,

$$N = \frac{250}{L^{\frac{2}{3}}},$$

or 250 revolutions for a stroke of one foot, or of 60 or 65 revolutions for 8 feet stroke. Twenty per cent higher speeds, both of piston and of rotation, may be taken to represent the practical limit, as set by considerations of prudence, for the most radical practice of the present time.

If the stroke be measured in inches,

$$N = \frac{1300}{l^{\frac{2}{3}}}, \text{ nearly;}$$

if in metres,

$$N = \frac{115}{L^{\frac{2}{3}}}, \text{ nearly.}$$

With a "detachable valve," or "drop cut-off," the number of revolutions is limited by that at which the valve-gear will operate with certainty and accuracy; and this seldom exceeds 100; although a speed of 125 revolutions has been attained, and special care may remove the working limit still further.

Standard speeds of rotation are sometimes adopted, as, for example, for naval engines. The British Admiralty accept the following, as maxima:

I. H. P.	Rev.	I. H. P.	Rev.
350	125	2000	90
500	120	3000	75
1000	110	5000	70
1500	100	6000	65

In designing marine machinery, in this matter, the efficiency of screw is often more influential than that of efficiency of engine. The friction-waste of engine increases at a much lower rate than that of the screw, and a restricted velocity is thus frequently found, on the whole, advisable. Speeds of 800 and 900 feet at sea, and 1000 feet per minute in locomotives, are now not uncommon.

Power and Speed Tables are constructed by every builder for his engines, which show what he expects to be their performance under the usual conditions of their operation. Such, for example, is the following for a series of standard simple engines, having "automatic" expansion-gear:

INDICATED POWER AT DIFFERENT SPEEDS AND PRESSURES.

CUTTING OFF STEAM AT ½ STROKE.

Size of Engine.	Constant.	Revolutions per minute.	Initial Steam-pressure.					
			50	60	70	80	90	100
7″ × 9″	.00175	300 340	13.1 14.8	15.7 17.8	18.3 20.8	21.0 23.8	23.6 26.7	26.2 29.7
8″ × 9″	.00229	300 340	17.1 19.4	20.6 23.3	24.0 27.3	27.4 31.1	30.9 35.0	34.3 38.9
8½″ × 10½″	.00302	270 310	20.3 23.4	24.4 28.0	28.5 32.7	32.6 37.4	36.6 42.1	40.7 46.3
9½″ × 10½″	.00376	270 310	25.3 29.1	30.4 34.9	35.5 40.7	40.6 46.6	45.6 52.4	50.7 58.2
10″ × 12″	.00476	250 290	29.7 34.5	35.7 41.4	41.3 48.3	47.6 55.2	53.5 62.1	59.5 69.0
11″ × 12″	.00576	250 290	36.0 41.7	43.2 50.1	50.4 58.4	57.6 66.8	64.8 75.1	72.0 83.5
12″ × 15″	.00857	210 250	44.9 53.5	53.9 64.2	62.9 74.9	71.9 85.7	80.9 96.4	89.9 107.1
13″ × 15″	.01005	210 250	52.7 62.8	63.3 75.3	73.8 87.9	84.4 100.5	95.0 113.0	105.5 125.5
14½″ × 17″	.01417	200 240	70.8 85.0	85.0 102.0	99.1 119.0	113.3 136.0	127.5 153.0	141.7 170.0
16″ × 17″	.01726	200 240	86.3 103.5	103.5 124.2	120.8 144.9	138.0 165.6	155.3 186.4	172.6 207.0

The "constant" is that quantity which being multiplied by the mean effective pressure and the revolutions per minute, the product is the horse-power.

Opposite is a speed and power table for a series of small direct-acting compound engines.

These tables may be taken as illustrating a common and fairly conservative practice.

4. **Simple and Compound Engines** and the various forms of the latter should usually be compared, by the processes which have been described, to ascertain whether the single, the double,

POWERS AND SPEEDS OF ENGINES.

CONDENSING ENGINES.

Indicated Horse-power.	Range of Initial Pressure. Pounds.	Diameter of High-pressure Cylinder. Inches.	Diameter of Low-pressure Cylinder. Inches.	Stroke. Inches.	Rev. per Min.	Piston Speed in Feet per Minute.	Floor-space occupied by Engine. Length. ft.	in.	Width. ft.	in.	Size of Steam-pipe. Inches.	Size of Exhaust-pipe. Inches.	Standard Fly-wheels. Diameter. Inches.	Width of Face. In.	Weight of Engine. Lbs.
60	90–110	8	14½	10½	275	481	10	7	4	5	2½	6	42	11½	6500
80	90–110	9	16	12	260	520	12	0	4	10	3½	7	48	12½	8500
130	90–110	11	19	15	230	575	14	7	5	6	4	8	60	14½	14500
200	90–110	13	23	17	210	595	16	7	7	4	5	10	80	18½	21500
250	90–110	15	26	17	200	566	17	8	8	0	5	12	84	18½	24000
325	130–150	13	23	17	200	566	16	10	9	1	6	10	86	25	30000
350	110–130	15	26	17	200	566	17	9	9	1	5	12	86	25	33000

NON-CONDENSING ENGINES.

Indicated Horse-power.	Range of Initial Pressure. Pounds.	Diameter of High-pressure Cylinder. Inches.	Diameter of Low-pressure Cylinder. Inches.	Stroke. Inches.	Rev. per Min.	Piston Speed in Feet per Minute.	Floor-space occupied by Engine. Length. ft.	in.	Width. ft.	in.	Size of Steam-pipe. Inches.	Size of Exhaust-pipe. Inches.	Standard Fly-wheels. Diameter. Inches.	Width of Face. In.	Weight of Engine. Lbs.
70	90–110	9½	14½	10½	275	481	10	7	4	5	3	6	42	11½	7000
90	90–110	10½	16	12	260	520	12	0	4	10	3½	7	48	12½	9000
150	90–110	13	19	15	230	575	14	7	5	6	5	8	60	14½	15500
250	90–100	15	23	17	210	505	16	7	7	4	5	10	80	18½	22500
275	90–100	16½	26	17	200	566	17	8	8	0	6	12	84	18½	25000
325	120–140	15	26	17	200	566	17	9	9	1	5	12	86	25	33000
350	120–140	15	23	17	200	566	16	10	9	1	5	10	86	25	31000

or other multiple-expansion type, is most likely, in the end, to prove of most advantage. In making this comparison, it is not only the costs of steam, of fuel, and of maintenance and operation that are to be studied; but, often, the distribution of work and the variations of rotating moments at the crank-shaft, and the effects of inertia may prove matters of sufficient importance to determine the choice of design or the proportions of details.

Where, as in marine work, a pair of engines is considered necessary, the compound engine costs nearly the same as the pair of simple engines, and weighs, including boilers, less, occupies the same or less space, and is wholly advantageous. The very large sizes and powers would have such enormous low-pressure cylinders, however, that it has become common to employ two each of one half the volume, thus securing the possibility of arranging three cranks at angles of 120°, getting a better distribution of pressures on pins and shaft, and more uniform turning moment. The triple-expansion engine permits a similarly satisfactory adjustment; when of very great power, its low-pressure cylinder may also be divided. In this case, and with quadruple-expansion engines, the cranks may be set in opposite pairs, thus securing balanced inertia-forces and still more uniform moments of rotation. The number and distribution of cylinders is thus settled by the judgment of the designer or by reference to ascertained data; and the several forms of engine elsewhere illustrated exemplify the conclusions of experienced designing engineers.

For moderate speeds of rotation, uniformity of turning moment is usually considered the main requirement; with high speeds of rotation, a balance of the inertia-effects of reciprocating parts is considered more essential; while, where costs of construction are to be given large weight, the "tandem" system of compounding is adopted. Either of these considerations may determine the general design of the engine.

Problems relating to the efficiency of the multicylinder engines are solved most simply by processes based on the method of Rankine, originally applied to the study of the ratio of expansion for highest efficiency of capital. The number of

cylinders or of grades of expansion being in all such cases settled by general experience and the judgment of the designing engineer, the best ratio of expansion and the best proportions of cylinders are readily determined for any given case by first obtaining the true curve of efficiency for the given class of engines, and then, knowing the probable back-pressure to be met with, either by custom or by taking it with reference to the best relation of initial to final pressure, and computing the constant and variable costs of operation, solving the problems, in their proper order, by a graphical construction which the Author has shown to be easily made.

In the design of the multiple-cylinder engine, the use of the steam-jacket is almost universal, although of less importance than in the simple engine with an equal total ratio of expansion, and is of less value as the speed of piston and the number of cylinders are increased. The working-barrel and the jacket are generally best cast separately. In adjusting the sizes and proportions of cylinders, it is advisable to adopt a three-crank system, if convenient, and to make the work done, the ranges of temperature, and the net initial stresses, as nearly similar as possible, a matter requiring care and some skill. The low-pressure cylinder demands especial care in the proportioning of its valves and ports, to reduce clearances and to secure straight and short passages. The expanded diagrams from the working engine, as compiled from the indicator-cards, should exhibit a smooth expansion-line and little lost work. The point of cut-off on the diagram is often five, or sometimes even ten, per cent earlier than the point of actual complete closure of the valve, in consequence of the "wire-drawing" taking place at that point with a slowly closing valve or a small port. The usual choice as to succession of three cranks gives the high-pressure the lead, places the low-pressure next, and the intermediate last.

The three-crank system gives the following advantages over the ordinary designs:

(1) Uniform stresses on the shaft;
(2) Balanced reciprocating stresses;
(3) Durability of wearing parts;

(4) Good adjustments of work and temperatures;
(5) Ease of movement at high speeds;
(6) Usually greater accessibility of parts;
(7) Interchangeability; facility of repair;
(8) A compact engine.

The cylinders can usually be so designed that the length of engine need not be much greater than with the engine having but two cranks.

The economical advantages of the detachable valve-gear, such as is usually desirable on the simple engine, are much less in the case of multicylinder engines; and it has been found, by experience, that excellent work may, in the latter class of machinery, be secured with the simplest forms of gear. Even the plain slide—preferably a piston-valve—without separate expansion-valve, is found effective on such engines. It is thus doubly advantageous to adopt the simplest possible type of valve-gear in the multiple-cylinder engines.

In choosing a valve-gear and in proportioning it, the endeavor should be to secure a symmetrical distribution at both ends of the cylinder, and, if possible, at all points of cut-off. With a single cylinder, it is sometimes advisable to use various forms of detachable or of independent cut-off gear; but, with multicylinder engines, the advantages of the single eccentric and its gear and the single valve, especially on engines of short stroke, may be found to fully equal those of the more elaborate kinds.

The higher forms of multiple-cylinder engine have now definitely displaced the lower, the two-cylinder compound, at sea, and seem likely ultimately to do so on land.

In comparing multiple-cylinder engines, it will be noted that the larger the number in series the less the ratio of expansion in each, for a given total expansion, and hence the less the difficulty in securing ample port-area and moderate valve-travel. Three- and four-crank engines balance well and have small friction as compared with those having two cranks, except the latter are set opposite, as in the Woolf engines, a sometimes objectionable arrangement, especially on shipboard. Unless

cost proves the ruling consideration, the quadruple-expansion engine, with four cranks, is, in this respect, desirable.

The three-crank compounded engine is less liable to give trouble by "getting caught on the centre" than even the engine with two cranks; but it may sometimes, especially if of short stroke, be found subject to this disadvantage; and a safe, handy, and quick-working steam reversing-gear is, for this reason, if for no other, always desirable.

The practical advantages of the multiple-cylinder engine are elsewhere fully discussed, as affecting the thermal efficiencies. The designer is also interested in the fact that defects of design, construction, or operation which result in leakage are less serious in the modern type; since the leakage from one may find use in the other cylinders and since the differences of pressure which determine the rate of loss are less.

According to Mr. Clark's statement: "The receiver-engine is an elastic system" of multiple-cylinder engine, permitting great latitude in adapting the receiver-pressure to the requirements of the succeeding cylinder without important diminution of its work; while the Woolf engine, losing pressure through the influence of its clearance-spaces, should have these intermediate volumes reduced to a minimum. Clearance is a much more serious element in the latter. The effect of 7 per cent clearance is shown by Clark, in a pair of cases taken for comparison, to be effective in the reduction of net work to the extent of 14 per cent in the Woolf and but 4½ per cent in the receiver-compound engine.*

5. **Compounding Engines**, apart from simply proportioning their cylinders, involves other no less essential considerations. The relative positions of cylinders; the relative motions of piston; the differences of steam distribution, and of turning moments at the shaft: all these, and other matters affecting the relative value and usefulness, and the several final efficiencies, must be considered and compared for the various designs under examination or proposed for adoption.

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* Steam-engine; vol. I. p. 443.

The two principal types of compound engine are best classified by the terms *direct transfer* and *indirect transfer*, although better known as the Woolf and the receiver engine or as engines having concordant and those having non-concordant strokes or crank-positions.

Thus a "tandem" compound engine is simple, of low first cost, durable, and in most respects satisfactory; while it has the disadvantage of a long steam-passage between the opposite outer ends of the pair of cylinders and that of giving no opportunity to secure smooth turning of the shaft with a light wheel; such as may be obtained where cranks can be set at angles of 90° or 120°; a "Woolf" or "McNaught" system, in which two cylinders are placed side by side, for example at one end of a beam, has the same disadvantages; a "Woolf system," in which the two pistons, in adjacent cylinders, have their motions in precisely opposite phases, attached each to its own crank, the two cranks set 180° apart, gives the best possible steam-distribution and balance; but is only equivalent, in rotative effect, to the engine with a single crank. The engines with cranks at other angles than 0° and 180°, or approximately so, must have receivers placed between their successive cylinders, in the series, and are hence classed as "receiver-engines."

Receiver-engines, as a class, must evidently be at some disadvantage in consequence of the existence of these intermediate receivers; which must waste some heat externally and must produce some malformation of the indicator-diagram by their interference with the ideal variation of pressures and volumes of working steam; such as would be illustrated either in a non-conducting, simple engine, or an engine of the second Woolf type, above described, if free from interior wastes. Yet they have the advantages of better distribution and smoother action in their turning moments at the shaft, and permit "reheating," re-superheating, the steam, if desired, at each transfer from cylinder to cylinder, as has been done at early dates by Corliss, Cowper, and Leavitt.

"Diagrammatic" illustrations of these various types of com-

pound engines are shown in the accompanying illustration.* The distribution of steam is shown by the arrows.

A, *B*, and *C* are of the primitive Woolf engine; *A* being the original; *B* is "tandem"; *C* is a "trunk" engine.

Another design, not here shown, is the annular engine of

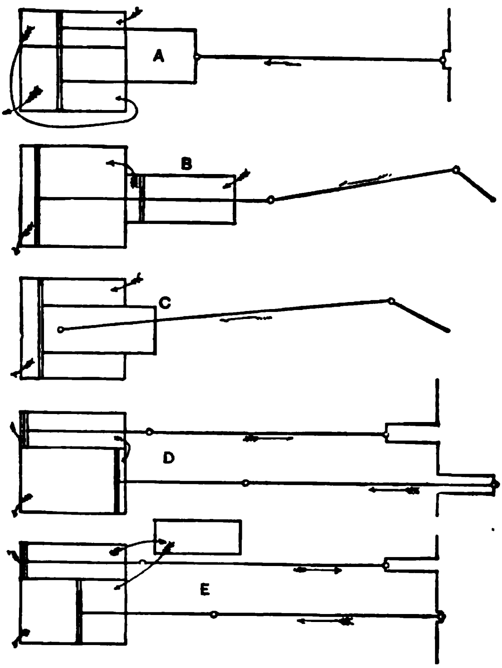

FIG. 1.—TYPES OF COMPOUND ENGINE.

Erastus Smith (1851, in SS. Buckeye State), in which the smaller cylinder is set inside the larger, and the pistons of both connected to one cross-head. *D* is the engine with cranks at 180°; and *E* is a "receiver-engine"; to which last class must usually be assigned all the multiple-cylinder engines with more than two in series, as well as all compound engines with two or

* See Whitham's "Steam-engine Design"; p. 128; and, also, Demoulin: "Machines à Triple et Quadruple Expansion."

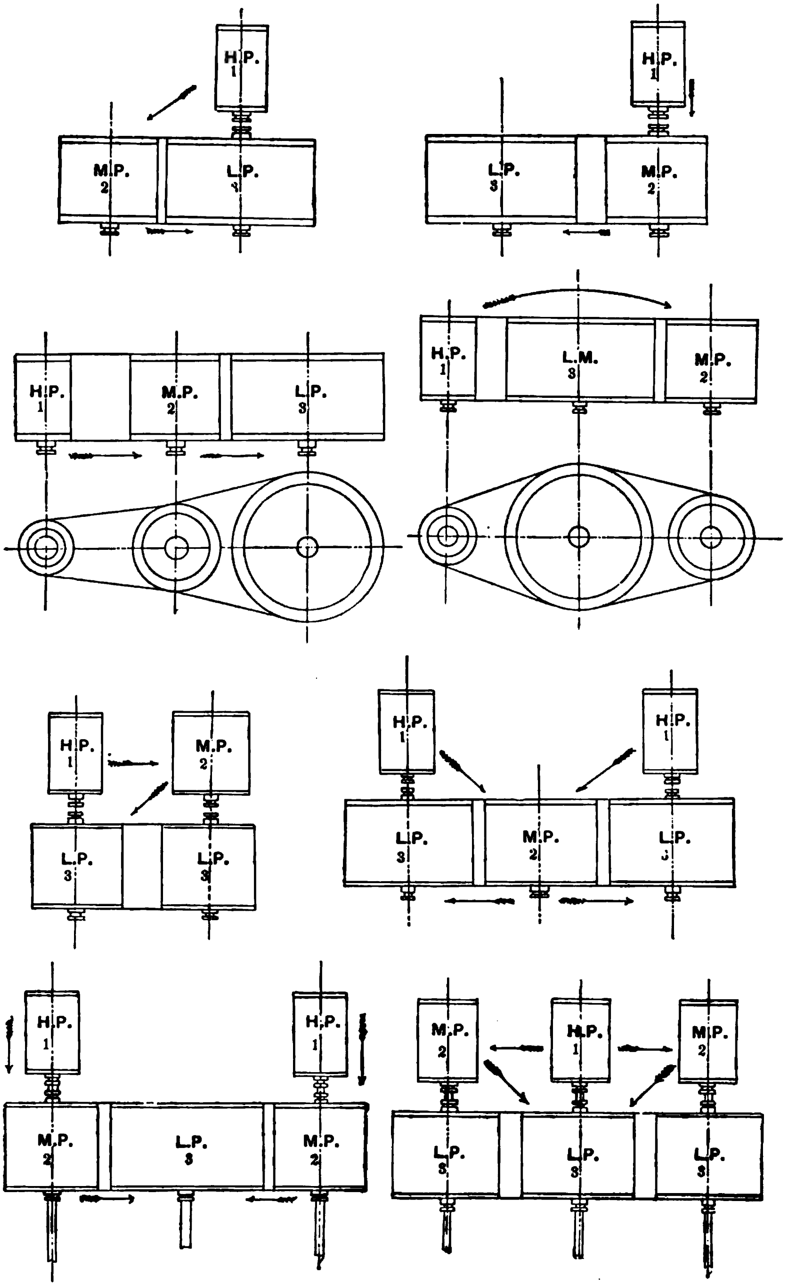

Fig. 2.—Grouping of Triple-expansion Cylinders.

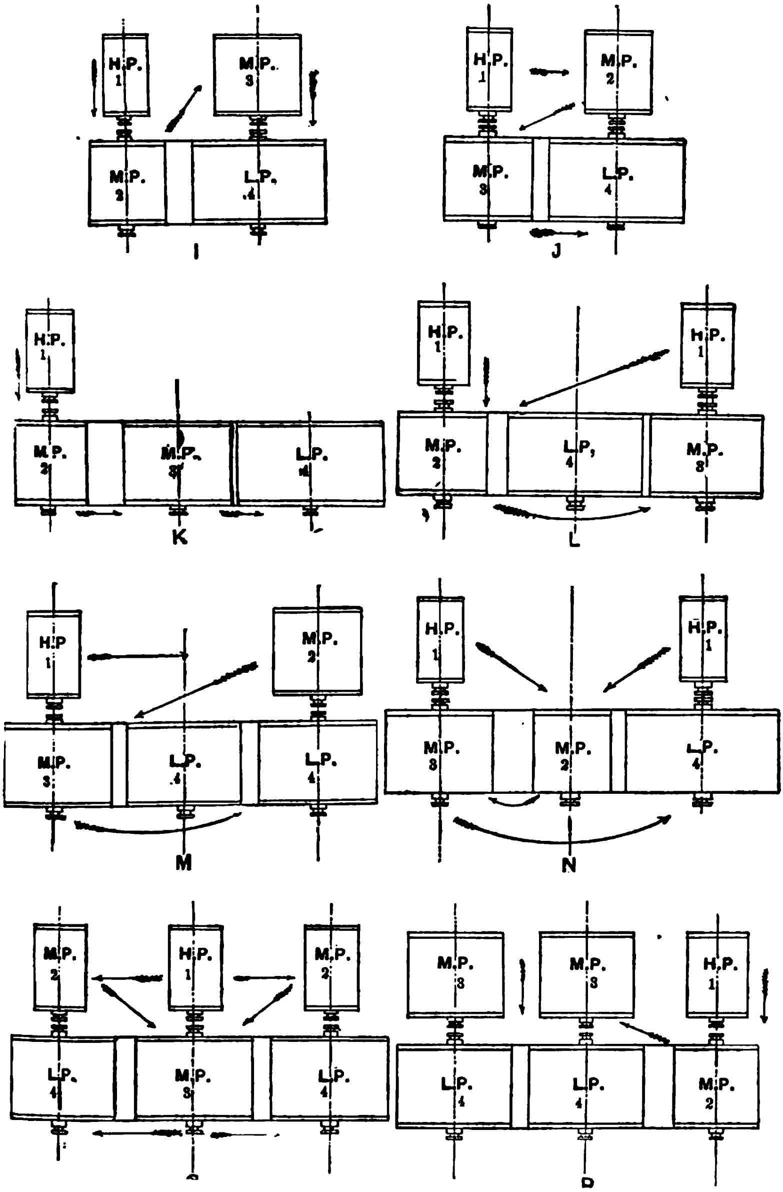

FIG. 3.—GROUPING OF QUADRUPLE-EXPANSION CYLINDERS.

more cylinders "in parallel", in series with a cylinder of higher pressure.

The receiver type, the Wolff, as distinguished from the Woolf according to Mallet,* has a decided advantage in its comparative independence of clearance volumes, and its permitting the receiver to act as a "separator" for the mixture of steam and water entering it, even when not used as, or in connection with, a "reheater."

"Triple-expansion Engines," called by French engineers "*machines à double cascade*," have been constructed occasionally since 1873; but have been a common and standard type in marine engineering since 1884. They are usually built with the three cylinders side by side, coupled to three equidistant cranks; i.e., set at angles of 120° with each other; this type having a better balance and less friction than the other forms. Quadruple-expansion engines, or "*machines à triple cascade*," have, with still higher pressures, succeeded, as a standard type, the preceding.

Figs. 3 and 4 illustrate a great variety of groupings of the cylinders of such engines. They are so marked as not to require detailed description here. Every designer has his own ideas relative to such arrangements, and many are patented, though often without basis in novelty.

The usual type of compound engine, especially at sea, has its pistons coupled to equidistant cranks, and a receiver of considerable volume is interposed between the cylinders to prevent that drop of pressure and that loss of energy inevitable where it is omitted. The larger this receiver the less is the economical working of the engine affected by the irregular steam-distribution. Could it be made of infinite size, no fluctuation of pressure could occur within it and no loss would be due to this phenomenon; but the larger the receiver the greater the loss of heat from it by conduction and radiation, the greater the weight, the volume, and the cost of the engine.

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* Compound Engines; Science Series, No. 10. D. Van Nostrand; 1884.

The receiver is, on these accounts, made as small as is consistent with smooth working and fairly satisfactory steam-distribution. The receiver is the exhaust-space for the high-pressure, and the steam-chest to the low-pressure, cylinder; within it a certain amount of intermediate expansion and fluctuation of pressure must always be submitted to. In the Woolf type this action may usually be ignored; in the one, the receiver is a clearance-space of real utility; in the latter, the clearances must be reduced to the least possible volume.

The elasticity of the system in the case of the receiver-engine is an advantage, in permitting a considerable range of adjustment of expansion and of work in the low-pressure cylinders; while, in the other type, the two cylinders must be very carefully proportioned to their prescribed steam-pressures, ratios of expansion and work, and greater losses follow any variations, in their operation, from the normal and intended working conditions. An adjustable expansion-gear is advantageously employed and an expansion-valve is always used, on the low-pressure cylinder of the receiver-engine, to measure into it the required quantity of steam and to secure satisfactory expansion; while, with the older engine, the high-pressure engine is the receiver, and the steam-measure, for the low-pressure cylinder, and no variation of its working ratio of expansion is practicable. Every variation of the ratio of expansion in the second cylinder of the receiver-engine obviously involves alteration of the receiver-pressure, increasing it with a higher and decreasing it with a lower ratio of expansion.

It follows from the above that, while the Woolf engine may be preferred when the work of the machine is constant and all conditions nearly invariable, as in the majority of pumping-engines; when the steam-pressures, the ratios of expansion, and the work of the engine are likely to be constantly and often considerably varied, as with marine engines, the receiver type is that to be chosen.

In the adjustment of relative proportions of cylinder and receiver some interesting problems arise which we have not

space here to consider in detail, but must be content with simple methods and usual cases.*

According to Ziese,† the following are the fundamental principles of the design of triple-expansion engines:

(1) The total power developed by an engine is independent of the relative volumes of the three cylinders and of the admission into the intermediate and low-pressure cylinders.

(2) Increased admission into the high-pressure cylinder increases to the same extent the work of the intermediate and low-pressure cylinder and *vice versa.*

(3) Decreased admission into the intermediate or low-pressure cylinders increases the back-pressure on the piston of the preceding cylinder and also increases the initial pressure in the intermediate or low-pressure cylinder.

(4) The final pressure in the high-pressure cylinder must be greater than, or, at least, equal to, the simultaneous pressure in its attached receiver; this also applies to the intermediate cylinder in its capacity as receiver for the low-pressure cylinder.

In all other respects the principles are analogous to those of compound engines.

The later builders have found that the steam-jacket, so essential to economy with the simple engine with high ratio of expansion, and also somewhat advantageous in the compound engine of two cylinders, may be omitted with much less disadvantage from the design of the triple-expansion engine; as may, also, the independent, or the "drop cut-off", valve so often introduced on single-cylinder stationary engines.

In designing the multiple-cylinder engines, we usually construct the proposed indicator-diagram as for a simple engine of similar power; then divide the work so as to apportion, as nearly as practicable, the same quantity to each of the several cylinders; next modify these parts in such manner as to allow fairly for anticipated losses of pressure and work in the actual

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* See The Engineer of Nov. 27, 1885, for illustrations of graphical processes devised by Professor R. H. Smith.

† Trans. St. Petersburg Polytechnic Society, 1886.

engine; and finally enlarge them proportionally, or alter their scale, so that, these losses being allowed, the net total work shall be that required.

6. Proportioning Cylinders in Series is a very simple matter when it is practicable to ignore the distorting effects of clearance, of friction of steam in pipes, and of the receiver. As has been already stated:

The number of subdivisions of expansion, and the minimum number of cylinders to be introduced in series, is finally settled by financial considerations. The fact that the loss by internal wastes is measured by that of one cylinder indicates that, as a matter of economy of heat, simply, there is no natural limit to the number, except that the losses by external conduction and radiation may finally more than compensate the gain by further complication.

The extent to which expansion may be economically carried in a single cylinder may be taken as ordinarily not less than two and a half expansions for unjacketed engines with wet steam and three or four for the better class of engines. The maximum total expansion-ratio thus becomes, for best effect, something like:

No. cyls....	1	2	3	4
r..........	2.5 to 3	6.25 to 9	16 to 27	40 to 81
p_1	25 to 30 lbs.	60 to 100 lbs.	120 to 300 lbs.	350 to 800 lbs.

The best ratios will often be found, for the better class of engines employing dry or slightly moist steam, to be, for unjacketed single cylinders, not far from one half the ratio of initial to back pressure, the latter including the friction of engine; and for multiple-cylinder engines of the highest class, using thoroughly dry or superheated and reheated steam, on the system adopted by Cowper, Corliss, and Leavitt, this best ratio may be raised economically, on the whole, to above two thirds the ratio of initial to back pressure.

It is safest to endeavor to find the real curve of efficiency for the class of engine considered, and use that curve in the solution of the problems of the efficiency of fluid, of efficiency

of engine, and of efficiency of plant. It thus becomes easy to ascertain the best total ratios for highest duty, for best financial results as designed, as for best commercial returns should the opportunity offer of utilizing more power than is at first anticipated.

Proportions of cylinders and relative ratios of expansion in the several cylinders of the multicylinder engine may thus be readily settled when the total ratio and the total power demanded are determined and exactly prescribed. It will be found that the ratio will be made, usually, not far from equality in the several cylinders, and the total ratio is then

$$r = r_1^n;$$

where n is the number of cylinders adopted, and r_1 the best ratio for one cylinder. It will, however, for best effect, on the whole, be probably advisable to adopt a compromise between the various modified and conflicting values prescribed by the conditions that the work, the effective initial pressures, and the areas into range of temperature, shall be as nearly equal in all cylinders as possible. To meet the first condition we must have such a ratio in each cylinder as shall make the work in each equal to the total net power of the engine divided by the number of cylinders in series; to meet the second condition we must make the initial pressure in each such that the total range of pressure may be equal to a common range in each multiplied by the number of cylinders; while to make the areas multiplied by range of temperature equal throughout the series, we must have varying differences of pressure, the high-pressure cylinder having the maximum range, and the low-pressure cylinder the minimum range, of pressure. The differences in this latter respect are, in engines using very high steam-pressure, quite considerable. Where the steam is dry, the speed of engine high, and the jacketing effective, this is a matter of less consequence than approximately uniform division of work and stresses on the crank-pins.

The proportions by volume, in the two-cylinder engine, v and V, of high- and low-pressure cylinders, are often given empirically, but the authorities disagree among themselves. Thus, Busley collates the following as proposed values of this ratio*:

$$\text{Grashof, } \frac{V}{v} = 0.85\sqrt{r};$$

$$\text{Hrabak, } \quad 0.90\sqrt{r};$$

$$\text{Werner, } \quad 1.00\sqrt{r};$$

$$\text{Rankine, } \quad 1.00\sqrt[3]{r^2}.$$

He considers the last the best for naval purposes, as giving a smaller high-pressure cylinder. Busley makes this ratio dependent upon the boiler-pressure thus:

p,	atmos.	4	6	7	8
	lbs. per sq. in.	60	90	105	120
$\frac{V}{v}$	. . . .	3	4	4.5	5.0

A correct method first determines the best ratio of expansion, all things considered, and then the best distribution of work and the proper corresponding ratios of expansion for each cylinder.

In the ideal design, there is no drop in pressure, or loss of work, in passing from the high- to the low-pressure cylinder, and the terminal pressure is equal, in each, to the back-pressure. In such case, the desired object would be accomplished, as already stated, by making the ratio of expansion the same for each cylinder, and equal to $\sqrt[n]{r}$.

For, in the two-cylinder compound, let

$U' =$ work in the first cylinder;
$U'' =$ " " " second "
$U =$ " " both (total);

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* Die Schiffsmaschine; Band 1. p. 364.

r' = ratio of expansion in the first cylinder;
r'' = " " " " " second " ;
r = ratio of total expansion;
p_1', p_2', p_3'; p_1'', p_2'', etc. = the several pressures;
v_1', v_2', v_3'; v_1'', v_2'', etc. = " volumes;

and assume hyperbolic expression. Then

$$U' + U'' = U = p_1'v_1'\left(\frac{1 + \log_e r'}{r'}\right) - p_2'v_2' +$$

$$p_1''v_1''\left(\frac{1 + \log_e r''}{r''}\right) - p_2''v_2''$$

$$= p_1'v_1'\left(\frac{1 + \log \bar{r}}{\bar{r}}\right) - p_2''v_2''.$$

But

$$p_2' = p_1'' = \frac{p_1'}{r'}; \; p_2'' = \frac{p_1''}{r''} = \frac{p_1'}{r'r''} = \frac{p_1}{r};$$

$$v_2' = r'v_1'; \; v_1'' = r'v_1' = v_2'; \; v_2'' = r_2''v_1'' = r'r''v_1';$$

and

$$U' + U'' = U = p_1'v_1'\left(\frac{1 + \log_e r'}{r'}\right) - p_1'v_1' +$$

$$p_1'v_1'\left(\frac{1 + \log_e r''}{r''}\right) - p_1'v_1'$$

$$= p_1'v_1'\left(\frac{1 + \log_e \bar{r}}{\bar{r}}\right) - p_1'v_1'.$$

Then, since $U' = U''$,

$$p_1'v_1'\left(\frac{1 + \log_e r'}{r'}\right) - p_1'v_1' = p_1'v_1'\left(\frac{1 + \log_e r''}{r''}\right) - p_1'v_1';$$

$$r' = r'', \quad \text{and} \quad r' = \sqrt{r}.$$

The initial pressure in each cylinder is the same, the length of stroke being, as is customary, the same; since the areas are in the proportion of $1 : r'$; and

$$p_1' - \frac{p_1'}{r'} = (r' - 1)\,p_1'; \quad p_1'' - \frac{p_1''}{r''} = p_1''(r'' - 1) = \frac{p_1'}{r'}(r' - 1).$$

Then, if the areas of piston be a' and a'',

$$p_1''a'' = \frac{p_1'}{r'} \times a'r' = p_1'a',$$

the two products measuring the net initial pressure in each cylinder.

The same process is evidently applicable to any number of cylinders in series.

Analytical methods commonly employed for proportioning cylinders are usually more or less closely related to the following, which was first published by Seaton*:

For Receiver Engines, two cylinders.

Let p_1 = absolute initial pressure of steam;
p_r = absolute receiver-pressure of steam;
p_2 = absolute condenser-pressure, or back-pressure in low-pressure cylinder;
R = ratio of cylinders;
r = total ratio of expansion;
r_1 = ratio of expansion in high-pressure cylinder;
r_2 = ratio of expansion in low-pressure cylinder;
p_m' = mean total pressure due to r_1 and p_1;
p_m'' = mean total pressure due to r_2 and p_r;
p_m = mean total pressure due to r and p_1;
p_e' = mean effective pressure in high-pressure cylinder $= p_m' - p_r$;
p_e'' = mean effective pressure in low-pressure cylinder $= p_m'' - p_2$.

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* Manual, pp. 95 *et seq.*

Also,

$$p_m = p_1\left(\frac{1 + \log_e r}{r}\right),$$

$$p_m' = p_1\left(\frac{1 + \log_e r_1}{r_1}\right),$$

and

$$p_m'' = p_r\left(\frac{1 + \log_e r_2}{r_2}\right).$$

But since the work is to be equally divided between the cylinders,

$$p_m - p_r = R\,(p_m'' - p_2). \quad . \quad . \quad . \quad . \quad . \quad (1)$$

And if there is no "drop," and the mean pressure in the high- is referred to the mean pressure in the low-pressure cylinder,

$$\frac{p_m' - p_r}{R} + p_m'' - p_2 = p_m - p_2,$$

which being combined with (1), we have

also,
$$\left.\begin{aligned} p_m'' - p_2 &= \frac{1}{2}(p_m - p_2); \\ p_m' - p_r &= \frac{R}{2}(p_m - p_2). \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If x denotes the efficiency of the system, and $(1 - x)$ is the loss due to "drop,"

also,
$$\left.\begin{aligned} p_m'' - p_2 &= \frac{x}{2}(p_m - p_2); \\ p_m' - p_r &= \frac{xR}{2}(p_m - p_2). \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

To find the *actual* mean pressures when there is a loss due to "drop," the value of x must be found: this may be done by substituting p_m' and p_m'' from the preceding formulæ; but an approximate value may be found by determining the value of p_r in (3); from the value thus found compute p_m'', referring the mean pressures of both cylinders to the low-pressure cylinder. If $(P_m - p_s)$ be the equivalent mean pressure thus found, then, approximately,

$$x = \frac{P_m - p_s}{p_m'' - p_s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The values of p_e, p'', and p_r are not precisely right, but are the theoretical mean pressures. In actual practice there are losses due to reduction of pressure in pipes and valves, steam-wastes in the engine, inefficient action in steam distribution and loss due to clearance, and other causes, often amounting to twenty per cent. The probable mean pressures, according to Seaton, may be found by multiplying those computed, as above, by the following factors:

Engine.	Factor.
(1) Simple expansive engine, special valve-gear, or with a separate cut-off valve, cylinders jacketed	0.94
(2) Simple expansive engine, having large ports, etc., and good ordinary valves, cylinders jacketed	0.90 to 0.92
(3) Simple expansive engines with the ordinary valves and gear, as in general practice, and unjacketed	0.80 to 0.85
(4) Compound engine with expansion-valve to h. p. cylinder; cylinders jacketed, large ports, etc............................	0.90 to 0.92
(5) Compound engines with ordinary slide-valves, cylinders jacketed, good ports, etc........	0.80 to 0.85

(6) Compound engines with early cut-offs in both cylinders, without jackets and expansion-valves, as in general practice in the merchant service .... 0.70 to 0.80

(7) Fast-running engines of the types and design usually fitted in war ships .... 0.60 to 0.80

Where a three-cylinder double-expansion engine is employed, the low-pressure cylinders are two in number and of equal volume, each one half the size of the single low-pressure cylinder displaced.

For Receiver Engines, with triple expansion, the process of computation may be substantially as in that of the two-cylinder compound, first assuming the ratio of volumes of low-pressure to high-pressure cylinder to be R, as before, and R' to be the ratio of the low-pressure to the intermediate. The effective mean pressures are, for the three cylinders, respectively,

$$p_m' - p''; \quad p_m'' - p'''; \quad p_m''' - p_s.$$

Neglecting loss due to the drop of pressure between cylinders,

$$p_m' - p'' = R(p_m''' - p_s); \quad p_m'' - p''' = R_1(p_m''' - p_s).$$

If P_m is the mean pressure due the total ratio of expansion,

$$P_m - p_s = p_m''' - p_s + \frac{p_m'' - p'''}{R_1} + \frac{p_m' - p''}{R};$$

whence

$$p_m''' - p_s = \frac{P_m - p_s}{3};$$

$$p_m'' - p''' = \tfrac{1}{3}R_1(P_m - p_s);$$

$$p_m' - p'' = \tfrac{1}{3}R(P_m - p_s).$$

The point of cut-off in each cylinder will be, respectively,

$$\frac{R}{r}; \quad R_1\frac{p'}{p''r}; \quad \frac{p'}{p'''r}.$$

In good engines of this kind, the wastes should not be more than a half or two thirds those in the preceding type.

Experience indicates that expansion-ratios exceeding 3 are probably rarely advisable in any one cylinder.

The practical, every day facts of engine operation must be kept in mind in the determination of proportions. Some designers would even prefer to make the usual systems of apportionment of work equally between two cylinders subordinate to the endeavor to secure a greater range of temperature in the small cylinder, in order to utilize the re-evaporated steam in the large cylinders.

Graphical methods are chosen, in some instances, the designer preferring to construct the diagram exhibiting the anticipated variations of actual volume and pressure of steam, and sometimes to still further approximate the real working conditions by introducing into the diagram the modifications due to inertia of reciprocating parts. He then adjusts the sizes of the cylinders so as to give, as nearly as may be, that uniformity of action of the steam upon the crank-shaft and running parts which he may judge likely to prove most satisfactory. This graphical system gives, generally,—as do nearly all other commonly practised methods,—larger cylinders, and a comparatively larger high-pressure cylinder, than that first described above, by, often, twenty per cent or more; since it may be made to include the influence of wastes, and of the back-pressures, as well. The diagram is made as exactly as possible like the diagram which the experience and judgment of the designer lead him to expect the engine actually to produce, and its area is then so divided as to assign to each cylinder its proper part. The mean pressure thus determined for each is assumed to be obtained in the working of the individual cylinder in such manner as to give it its share of the total work; the assigned horse-power then determines the area of its piston, thus:

$$A = \frac{a \times H.P.}{V \times p_m}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

in which A is the piston-area; $H.P.$ is the power demanded of the given cylinder; V is the speed of piston; a is the work of a horse-power in the same unit of time, as 33,000 foot-pounds per minute, in British measures; and p_m is the mean pressure found, from the diagram, for that cylinder. If the corners of the diagram-sections are rounded off as in the actual indicator-card, and the intermediate drop of pressure shown, the mean pressure, and consequently the size of cylinder, will be obtained with accuracy.*

To prevent a "drop" in a receiver engine, as already seen, a ratio of expansion, or point of cut-off, must be found such that the compression in the small cylinder shall carry the steam-pressure up to that at which the large cylinder began to take its steam. This point may be easily found, assuming hyperbolic expansion, thus:

Let p_r = the required pressure;
R = ratio of volume of receiver to that of small cylinder;
V_l = ratio of volume of large to that of small cylinder;
r = " " expansion demanded in large cylinder.

Then the total volume of the steam at the opening of the exhaust-valve of the small cylinder is $1 + \cancel{V_l}$ R, and at the cut-off $\frac{1}{r}$, $R + 1 - \frac{1}{r} + \frac{1}{r} V_l$; and the compression then gives, in the small cylinder, a reduction of volume of the compressed steam from $R + 1 - \frac{1}{r}$ to R, if we neglect clearance. The pressure to be determined is

$$p_r = \frac{p_r(R+1)}{R + 1 + \frac{1}{r}(V_l - 1)};$$

* See Sennett; pp. 496, 497.

while the same pressure must be equal to

$$p_r = \frac{p_r(R+1)}{R+1+\frac{1}{r}(V_l - 1)} \times \frac{R+1-\frac{1}{r}}{R};$$

and

$$r = \frac{RV_l + 1}{R+1};$$

which gives, for $r = 2\frac{1}{2}$, $R = 1$, and $V_l = 4$, for example.

It will be found that ordinary practice and proportions will give a small receiver-volume for cranks at right angles, and usually in the proportion of about 1 to 2.

For the case of the compound locomotive, as usually designed, with cranks at right angles and intermediate receiver, the following are the results of varying receiver-volumes:

Vol. Rec. to h. p. cyl.	r	p_1	p_2	p_3	p_r	p_4	p_5
1.0	2	160	80	107	91.8	80	40
1.5	2	160	80	100	88.9	80	40
2.0	2	160	80	96	87.4	80	40
1.0	2.5	160	64	96	80.2	64	26
2.0	2.5	160	64	83	74	64	26

where r is the ratio of expansion for both cylinders; and the pressures, in order, absolute initial; terminal in the h. p. cylinder; maximum and mean receiver-pressures, and in the l. p. cylinder up to point of cut-off; pressure at cut-off and at end of expansion in the latter, hyperbolic expansion being assumed throughout. The most usual values of r in practice approximate 2, but occasionally are not far from 3.

Fig. 4 exhibits the best proportions of intermediate receiver for various proportions of the cylinders, where their relative volumes vary from $2\frac{1}{2}$ to 5, and for two types of engine, the one having cranks at 90°, the other at 180. R is the ratio

in the large and r that in the small cylinder; and V_1, V_2 are their volumes.

The term "Wolff compound" is now very generally applied to all engines working with receiver, whether of the

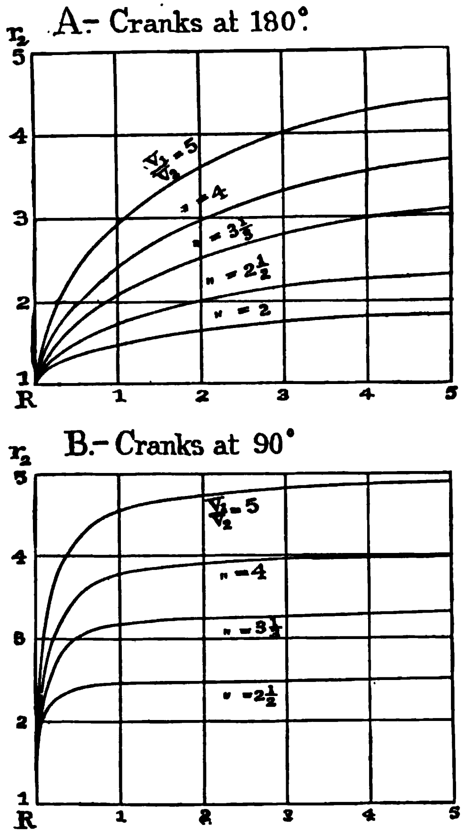

FIG. 4.—RECEIVER-VOLUMES AND EXPANSION-RATIOS.

original Wolff form or not. In the receiver engine, the connecting pipes and the valve-chests together frequently give all the capacity desired, and no separate vessel is used. This receiver-space, in some form, is to be provided with even the

Woolf arrangement of cylinders when the high-pressure cylinder is to have a variable expansion-gear. As in all other cases, the low-pressure cut-off is adjusted, in normal working, so that little or no drop of pressure may take place between the cylinders, in the receiver.

The use of a provisional diagram, as the best and simplest method of proportioning a multiple-cylinder engine, is illustrated by the following sketch and the accompanying outline.* The ratios of volume of the several cylinders should first be settled and the clearances determined, by reference to experience with the same class of machine, if possible. The curve of Mariotte, *AB*, is first laid down, as shown, Fig. 5, as approximately representing the ideal expansion-line, assuming the total volume of steam received and the total ratio of expansion known in advance. The mean pressure is readily computed for this case, and the relation of its value and of the volumes and pressures of the real engine are approximately known by reference to engines in operation under the proposed working conditions.

The coefficients given by Seaton and other authorities are thus obtained and used. The approximate diagram for the real engine may thus be obtained and drawn within the ideal curve, the successive diagrams of the several cylinders being distinguished and allowances made for the falling steam-line, *ab*, the altered expanding curve, *bcd*, the drop of pressure, *fa*, between the diagrams, and the clearances and compression-lines as shown. A little readjustment fixes the best distributions of work, pressures, and ratios of expansion and the then measured mean pressures give the required volumes of the cylinder; which volumes, divided by the lengths of piston-stroke, give the areas of piston. No account is here taken of the probable wastes of steam and heat by cylinder-condensation and leakage. They only affect the computations of dimensions

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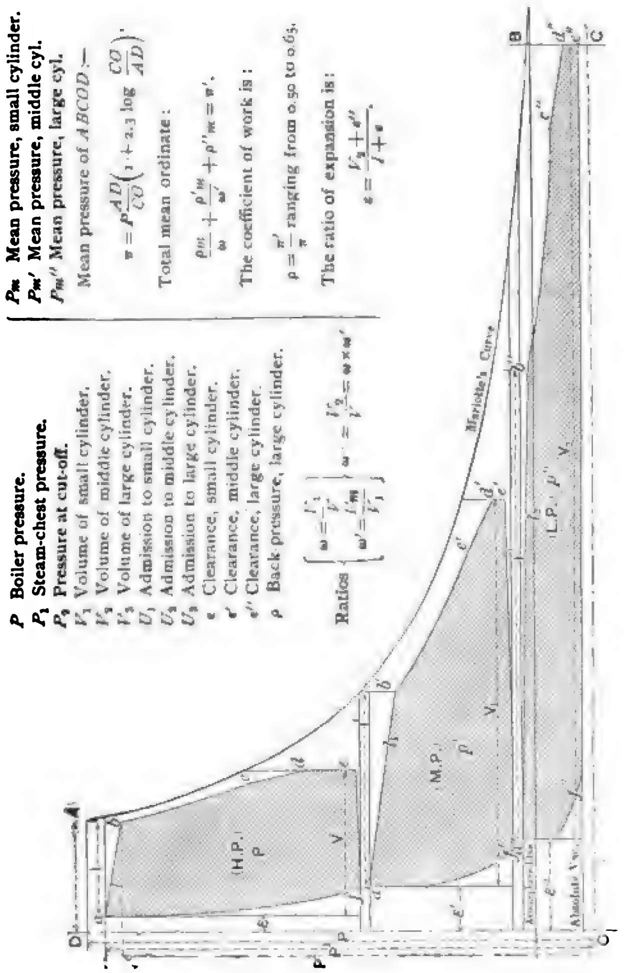
* See Demoulin: "Les Machines à Vapeur, à Triple et Quadruple Expansion."

of boiler by adding the known or estimated percentage to the quantity of steam demanded.



Including "dead-space," $r = \frac{V_3}{l} = \frac{V_3}{V} \times \frac{1}{i} = \frac{\omega}{i}$; i being the cut-off in the h. p. cylinder $\frac{l}{V}$.

Fig. 5.—Provisional Diagram (*Triple Expansion*).

7. The Distribution of Steam and of Work being determined, the design of each cylinder and of the various details of the engine may be proceeded with.

Since the performance of an engine cannot be predicted

with exactness and certainty, it is necessary to make assumptions based on reliable experiments.

In designing, the quantities to be determined are:

The diameter of cylinder and length of stroke of an engine to furnish given horse-power, the following being given:

Pressure of steam.

Ratio of expansion.

Revolutions or piston-speed.

If the mean effective pressure of the steam at each stroke can be determined, or if the indicator-diagram can be laid

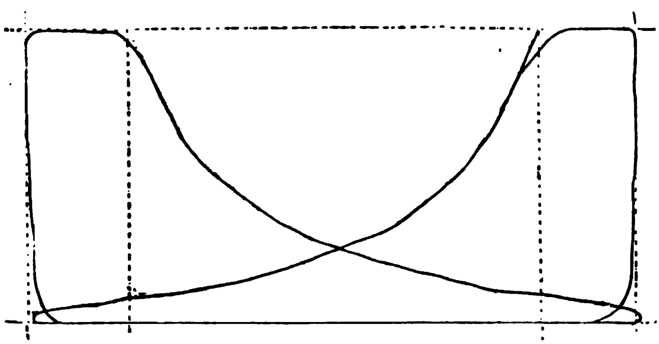


FIG. 6.—ENGINE WITH DETACHABLE VALVE-GEAR—30" × 48"; Steam, 84 lbs.; Speed, 56.

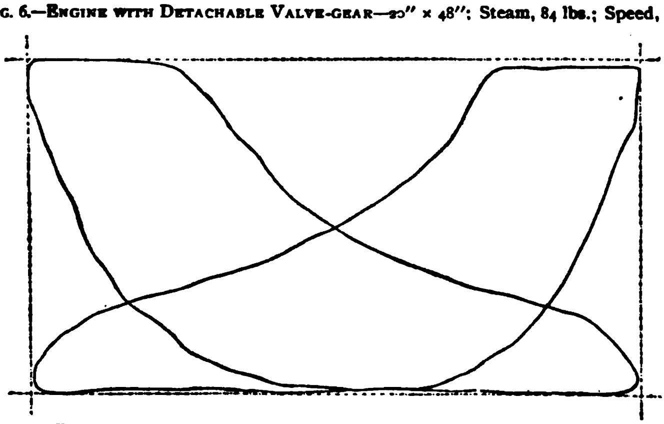


FIG. 7.—AUTOMATIC ENGINE—10" × 14"; Steam, 90 lbs.; Speed, 240.

down in advance of construction, these dimensions can be obtained. The diagram in Fig. 6 is given by a Corliss engine

of satisfactory design; while the next example, Fig. 7, is such as is produced by an equally excellent representative "high-speed automatic engine." Each illustrates well the characteristics of these two extreme types of engine; the one having a moderate speed of rotation and a quick-working cut-off valve, large expansion, and no compression; the other precisely the opposite, a high-speed engine with positive valve-gear, less expansion, and requiring much larger compression to fill its clearance-spaces on the return-stroke. Either may be taken as the representative indicator-diagram for the case to be assumed.*

In laying down a diagram to be used as a basis of the work, the ideal diagram is to be first drawn, with its steam-line of constant pressure, its parabolic expansion-line and rectangular lower portion; then the figure is modified to give it the form considered to be the nearest practicable approximation to the ideal, by rounding its corners and adjusting its compression-line. This modified diagram is that upon which the design is based.

It is to be remembered that considerable compression—and hence comparatively large clearance and port spaces—is desirable to the engine of high speed, to secure smooth running. It is this which gives the characteristic difference in the two diagrams just referred to.

OX (*A*, *B*, *C*, *D*), Fig. 8, is the zero line of absolute pressures.

1. The initial pressure p_1 diminishes to p at the point of cut-off, *B*, where the steam-valve closes.

It is not known how much the pressure will fall, but a difference of five pounds between p_1 and p will be ample allowance. An equivalent line, *HA*, can be drawn, representing mean pressure up to point of cut-off $= \dfrac{p_1 + p}{2}$.

2. When the steam-valve closes, the steam in the cylinder expands to f, and the exhaust-valve opens.

* For more of detail, see Du Bois' Weisbach; vol. II. part II. pp. 468 et seq.; N. Y., J. Wiley & Sons.

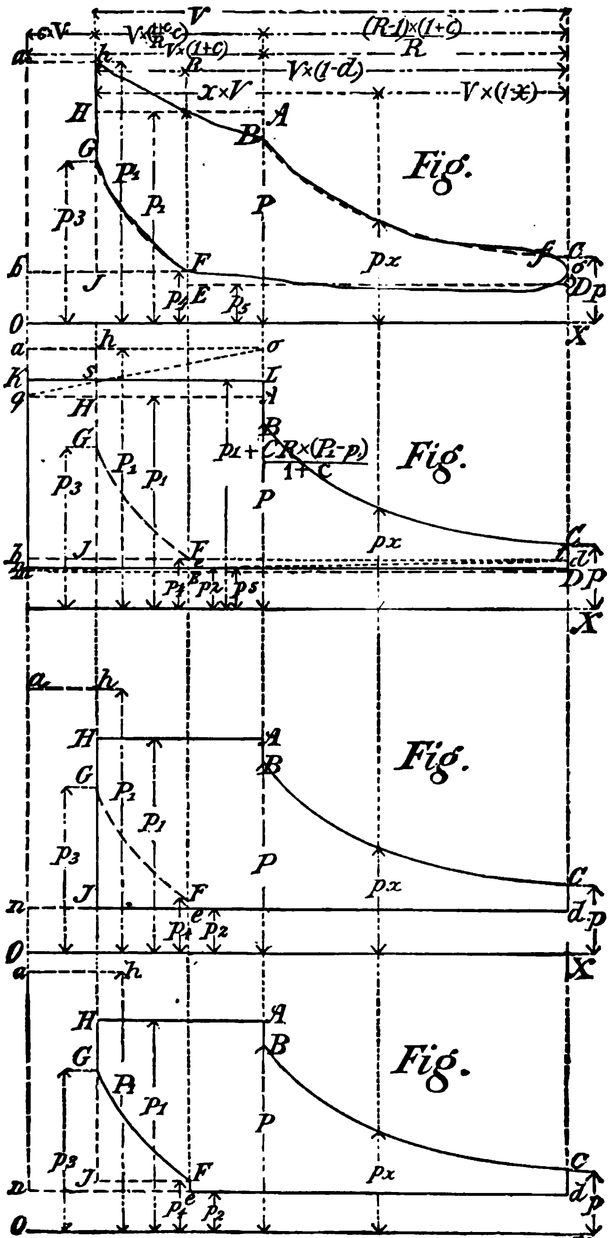


FIG. 8.—REPRESENTATIVE DIAGRAMS.

If the ratio of expansion is known, the curve of expansion can be laid down. The points at which cut-off, release, and cushion commence can be fixed at pleasure. But the fraction of stroke completed up to point of cut-off is only the *apparent cut-off*, and the *real cut-off* is

$$\frac{\text{stroke to cut-off} + c}{\text{stroke of piston} + c};$$

in which c is the clearance measured in the same units as the stroke of piston. This is found to vary between 5 and 10 per cent. The clearance can be assumed somewhere between these figures.

3. On the return-stroke, the exhaust-valve remains open, with variable back-pressure, which increases to F, where the exhaust-valve closes.

With good exhaust ports the back-pressure should be usually between one and two pounds per square inch of piston-area in non-condensing, and between four and five pounds (above zero) in condensing engines.

4. When the exhaust-valve closes, the steam is compressed, and at G steam is again admitted to the cylinder.

The compression-curve may be assumed to follow the same law as the curve of expansion. The most economical ratio of compression is, as has been elsewhere shown, greater than the ratio of expansion.

The notation adopted is as follows:

A = area of piston, in square inches;
S = length of stroke, in feet;
s = distance from commencement of stroke to point of release, in feet;
D = distance, in feet, from commencement of stroke to point of cut-off;
c = fraction of clearance;
P_1 = initial pressure of steam;
P = pressure of steam at point of cut-off;
p_1 = mean pressure up to point of cut-off, clearance-space not included;

p_3 = mean back-pressure, exclusive of cushion;
p_4 = back-pressure at point where cushion commences;
t = temperature of steam, on Fahrenheit scale at pressure P;
t_x = temperature of steam, on Fahrenheit scale, at pressure px;
w_x = weight, in pounds, of a cubic foot of steam, at pressure x.

All pressures, when not otherwise stated, are total pressures (above a vacuum), in pounds per square inch.

The use of the tables is illustrated by the following examples:

(1) If steam of 80 pounds pressure per square inch is allowed to expand to 25 pounds, to find the pressures at three intermediate points and the mean pressure, supposing that the pressure varies inversely as the $\frac{17}{16}$ power of the volume.

By formula, the real cut-off is $\left(\frac{25}{80}\right)^{\frac{16}{17}} = 0.31^{\frac{16}{17}}$, which is found in column 3 of Table V to be 0.3321, and the ratio of expansion is 3.01. Hence the ratios of expansion for the intermediate points are 1.5, 2, and 2.5. Entering Table V with these quantities in column 2, the pressures at the points are, from column 4:

1st point,	pressure		$= 80 \times 0.6534 = 52.3$;
2d	"	"	$= 80 \times 0.4788 = 38.3$;
3d	"	"	$= 80 \times 0.3777 = 30.2$.

The mean pressure is $80 \times 0.5276 = 42.2$ pounds per square inch.

(2) A cylinder is 24 inches in diameter, and the stroke 4 feet. The clearance is $\frac{8}{10}$ of a cubic foot. The engine makes 60 revolutions per minute, the steam cut off at 16 inches, and cushion commences when the piston is within 12 inches of the end of stroke. The steam follows, approximately, the law of Mariotte.

Initial pressure, P_1, 70; pressure at point of cut-off, P, 66; mean back-pressure, 5: back-pressure where cushion commences, p_4, 6. To find the indicated power:

It is found that the space swept through by the piston, per stroke, is

$$V = 4 \times 0.7954 \times 4 = \cancel{2}2.5664 \text{ cubic feet.}$$

The fraction of clearance, at each end of cylinder, is

$$C = \frac{12.5\cancel{6}64}{\cancel{0}.8} = 0.064.$$

The apparent and real cut-off are

$$\frac{1}{r} = \frac{16}{48} = 0.33; \quad \frac{1}{R} = 0.37.$$

The real ratio of expansion,

$$R = 2.703.$$

The terminal pressure,

$$p = 66 \times 0.37 = 24.4 \text{ lbs. per square inch.}$$

The mean pressure up to point of cut-off,

$$p_1 = \frac{70 + 66}{2} = 68 \text{ lbs. per square inch.}$$

The mean pressure up to point of cut-off, clearance-space included,

$$P_i = 68 + \frac{0.064 \times 2.703 \times (70 - 68)}{1.064} = 68.3 \text{ lbs. per square inch.}$$

The mean effective pressure, uncorrected,

$$P_m = \frac{68.3}{2.703} + 66 \times 0.3679 - 5 = 44.4 \text{ lbs. per square inch.}$$

The mean effective pressure, corrected,

$$p_m = 44.4 - 0.064 \times (70 - 44.4) = 42.8 \text{ lbs. per square inch.}$$

The fraction of stroke uncompleted when cushion commences,

$$d = \frac{12}{48} = 0.25.$$

The ratio of compression, for cushion, is

$$\frac{0.25 + 0.064}{0.064} = 4.9.$$

The final cushion pressure,

$$p_2 = 6 \times 4.9 = 29.4 \text{ lbs. per square inch.}$$

The mean cushion pressure,

$$p_c = 29.4 \times 0.4024 = 11.8 \text{ lbs. per square inch.}$$

The mean effective pressure, corrected,

$$P_c = 42.8 - 0.064 \times 4.9 \times (11.8 - 6) = 41.4 \text{ lbs. per sq. inch.}$$

Hence,

$$I.H.P = \frac{41.4 \times (24)^2 \times 0.7854 \times 480}{33{,}000} = 272,$$

and would reduce the indicated horse-power to

$$\frac{41.4 \times 447.58 \times 480}{33{,}000} = 270,$$

and the net horse-power to

$$270 \times 0.85 = 229.$$

(3) Suppose that p_1 is 35 pounds per square inch, and that the engine is to exert 350 net horse-power, with a piston speed of 500 feet a minute, the net horse-power being 82 per cent of the indicated.

Then p_e is

$$35 \times 0.82 = 28.7 \text{ lbs. per square inch,}$$

and each inch of piston area will give

$$\frac{28.7 \times 500}{33,000} = 0.43485 \text{ net horse-power,}$$

and the effective area required will be

$$\frac{350}{0.43485} = 804.88 \text{ square inches,}$$

or a cylinder 32 inches in diameter; and if a piston-rod $4\frac{1}{2}$ inches in diameter is required, the total area should be

$$\frac{2 \times 804.88 + 15.90}{2} = 812.83 \text{ square inches,}$$

and the corresponding diameter is

$$\sqrt{\frac{812.83}{0.7854}} = 32.17 \text{ inches.}$$

If the speed is fixed at 50 revolutions per minute, the piston must move

$$\frac{500}{50} = 10 \text{ feet per revolution,}$$

and the stroke is

$$\frac{12}{2} = 5 \text{ feet.}$$

With the "compound" or other multiple-cylinder engine, precisely the same processes apply, the diagram employed being that appropriate to the cylinder under consideration.

In the following discussions, the "line of connection" will be followed in taking up the more important parts in their

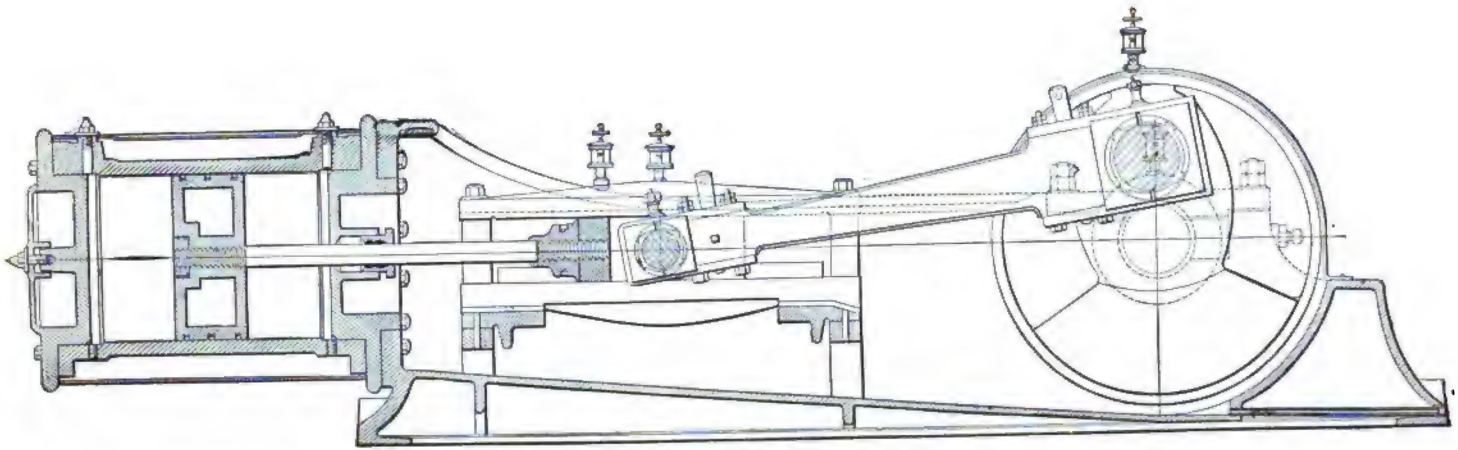


FIG. 9.—LINE OF CONNECTION.

order, and it will be assumed that these parts have the general form and relation illustrated, for example, by the above excellent typical case, a general design by Mr. P. H. Ball, in which simplicity, stability, and compactness are sought for the case of a high-speed engine. (Fig. 9.)

The sketch may be studied as illustrative, also, of the methods of connection, as well as of a good general grouping.

8. Work and Steam in Compound Engines, and in others having cylinders in series, are distributed variously, as the method of their combination may make necessary. The manner in which this is determined by the type of engine selected has already been indicated in a general way. Their diagrams of energy for the ideal case, and indicator-diagrams of the real engine, which represent that distribution, are commonly as shown below. The departure of the curves of the real engine from the straight lines and smooth expansion-curve of the ideal diagram measure the kinematic defects of the engine and their modification of the distribution of steam.*

Compound-engine Diagrams, as produced by the steam-engine indicator, often differ, even more than with the simple engine, from that ideal "card" which would be given were the expansion precisely as intended, and the engine free from defect, as in clearance-spaces and port-resistances. The best compound engines show considerable loss, as has been seen, in these ways, and also in that drop of pressure between high- and low-pressure cylinders which often constitutes a very sensible source of waste of heat, steam, and fuel. Where the Wolff system is adopted, however, if the load be constant, and the machine well proportioned to its work, and if the "dead-spaces" can be made small, the approximation of the actual to the ideal

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* Mr. J. McFarland Gray states that he finds the mean effective pressure in the compound engine, reduced to the low-pressure cylinder, to be very approximately

$$p_m = \sqrt{6p_1};$$

where p_1 is approximately, or exactly, boiler pressure. This expression may be employed in laying down these typical diagrams.

card may be very close, as is illustrated by the accompanying pair of diagrams from a pumping-engine of this character.*

The action of the steam and its variations of pressure are here seen throughout the cycle to be precisely similar to that in a simple engine. Large steam-ports and a good expansion-gear bring the steam-line close up to that of boiler-pressure; a well-jacketed cylinder allows the expansion-line to follow closely that laid down for the ideal engine; short and free ports between the two cylinders give an exhaust from the high-pressure and a supply to the low-pressure cylinder which are nearly coincident; and the two cards would, if reduced to a single diagram, exhibit a very close approximation to that which would have been constructed as the ideal diagram of this class of engine. When these points are not well attended to, variations of twenty and even thirty or forty per cent may be observed between the computed power, as based on the designer's indicator-cards, and the actual work of the engines under their ordinary conditions of operation.

It has long been known that there may be determined a certain definite ratio of expansion in the high-pressure cylinder of a multiple-expansion engine, such that, at that ratio, the mean pressure on its piston is a maximum.†

Thus, assuming hyperbolic expansion, and taking

v_1 = volume of the h.-p. cylinder;
v_2 = " " " next "
$\frac{v_1}{r_1}$ = cut-off in " h.-p. "
r_1 = ratio of expansion h.-p. cylinder;
p_1 = initial pressure in " "
$p_2 = p_b$ = pressure at end of its stroke;
$1 \div r_2$ = cut-off in the l.-p. cylinder;—

* For a full and clear treatment of this subject in its minor details, see D. K. Clark's Manual, p. 849 *et seq.*, or his Treatise on the Steam-engine; 1889–90.

† Trans. Inst. Nav. Architects of G. B.; vol. XIX. p. 205.

$$p_2 = p_1 \frac{v_1 \div r_1}{v_2 \div r_2} = \frac{p}{r_1 r_2};$$

while the work done per stroke is

$$U = p_1 v_1 \frac{1 + \log_e r_1}{r_1} - p_2 v_2 = p_1 v_1 \left(\frac{1 + \log_e r_1}{r_1} - \frac{1}{r_1 r_2} \right);$$

which is a maximum when, taking r_1 as variable,

$$\log_e r = \frac{1}{r_2}.$$

When $r_2 = 1$, as in the ordinary engine, we have

$$r_1 = e = 2.718.$$

Thus, for example, with cylinders having volumes as 4 : 1; $r_2 = 2$; steam at 80 lbs. pressure, $r = 1.65$, and $p_m = 45.4$ lbs. per square inch.

This value is somewhat modified by the presence of the intermediate passages between the cylinders, a drop occurring in the pressure at the instant of opening the exhaust from the small cylinder; but this drop is less as those passages are larger; and if forming an intermediate reservoir, as is sometimes the case where "reheating" between the cylinders is practised, this loss and the corresponding reduction in the mean pressure obtained, in work done, and in the actual total ratio of expansion, is sometimes quite unimportant compared with the gain by that process. A common value for the reduction of total expansion is not far from 20 per cent, rising to one third with small reservoirs and falling to a lower figure with larger spaces. The loss of work may usually be neglected.

The receiver type of engine with equidistant cranks and intermediate reservoirs is less seriously affected by intermediate spaces. The reduction of pressure and the loss of total expansion is but about 10 per cent, where the receiver-space is equal to the volume of the smaller cylinder, and falls to less than 5, in usual cases, when the receiver is as large as the larger

cylinder; losses which may be easily approximately estimated and allowed for in any case.

In the next illustration, Fig. 10, from Mr. Porter's report, the natural form of the expansion-line, in the single cylinder, having the capacity here observed in the low-pressure engine, would be that shown by one or other of the two dotted lines, accordingly as the expansion approached more or less closely the hyperbolic form. The initial volume is *AB*, and the press-

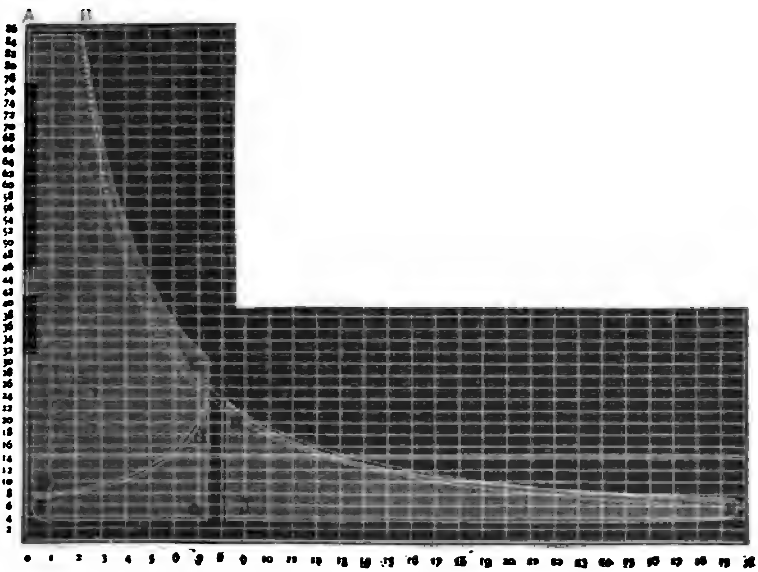

FIG. 10.—COMPOUND-ENGINE DIAGRAMS.

ure as shown on the vertical scale; while the gradual loss of pressure with increase of volume is shown by the two scales as the line progresses toward the right to its terminal point at *I*. The deviation from the dotted line of the actual expansion-line between *B* and *C* illustrates the gain of weight and pressure due to the progressing re-evaporation of steam originally condensed in the cylinder at the opening of the steam-valve, and to the admission of the fluid into the colder cylinder. Here expansion occurs from the initial pressure and volume at *B*

down to the terminal point C in the high-pressure, and from C or H to I in the low-pressure, cylinder. The indicator-diagrams actually obtained are $ABCD$ and EFG, the latter being the equivalent in the low-pressure cylinder of the card HIJ, which would have been produced had the high-pressure cylinder been given sufficient length to permit the completion of the expansion in that cylinder. The variation of the full line,

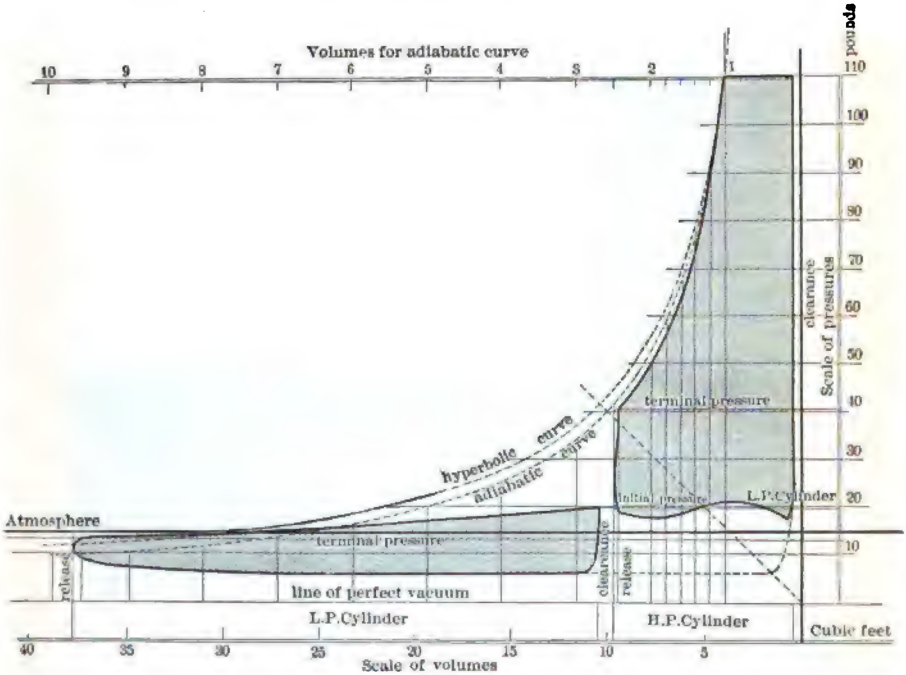

FIG. 11.—ACTUAL AND IDEAL DIAGRAMS; COMPOUND ENGINE.

representing the real diagram, from the ideal dotted expansion-line, is indicative of the fluctuations of pressure produced by the condensation and re-evaporations taking place as expansion progresses in the metallic chamber serving as working cylinder.

The succeeding figure illustrates the visible differences between the diagrams actually taken from the two cylinders of a compound engine—in this case a "Reynolds-Corliss"—and the ideal combined card.

This next diagram, from an engine of similar class with the preceding, and published by its designer, exhibits at once the method of reducing the actual indicator-diagrams to the combined form, and the variations from the ideal expansion-line due to imperfections of the engine as a work of human art. Pressures are measured in pounds on the square inch, and volumes in cubic feet, actual capacities of cylinder being given. As shown on the diagram, about $3\frac{1}{2}$ cubic feet of steam enter the high-pressure cylinder, each stroke, at a pressure of 110

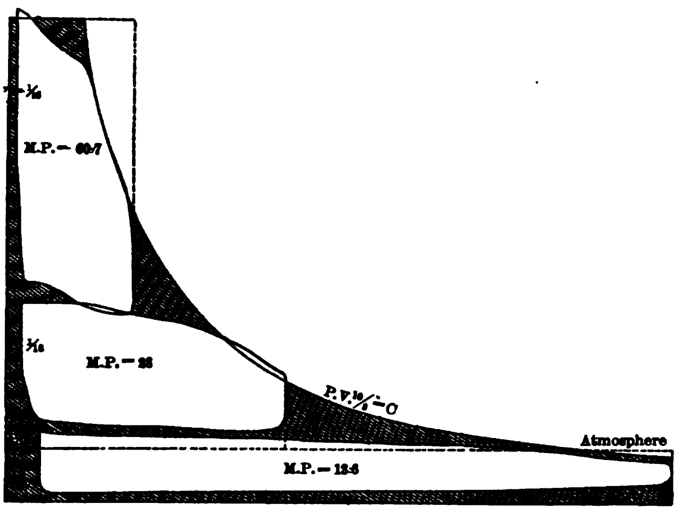

FIG. 12.—TRIPLE-EXPANSION DIAGRAM.

pounds per square inch above vacuum; it expands nearly adiabatically to $9\frac{1}{4}$ cubic feet, is then transferred to the low-pressure, dropping from the terminal pressure, 40 pounds in the high-pressure cylinder, to 20 in the low-pressure, and then expanding in the latter down to about 12 pounds when it passes into the condenser, the back-pressure thus becoming not far from an average of 6 pounds. The two indicator-diagrams are shown by the "hatched" spaces; the ideal diagram incloses both, its outline being the dotted lines. The very considerable space measuring the difference of the two areas is a gauge of the imperfection of the cycle. The departure of the actual line

from the two ideal expansion curves, and the fact that the former lies within both the latter, indicate that the jacket does not supply heat enough to compensate the condensation of the expanding fluid; far less enough to retain its temperature constant or to continuously superheat it.

The discordant fluctuations of similar lines in the two indicator-diagrams exhibit the effect of non-synchronous motion of the two cylinders.

The accompanying illustration (Fig. 12) exhibits the proportions of the diagrams taken from a triple-expansion engine, drawn to common volume and pressure scales, and placed under the Mariotte line. The engine has cylinders having the ratios 1 : 2.25 : 2.42, and the total ratio of expansion is 8, the cut-off in the several cylinders being set at 1.47, 1.3, 1.3. An advantage of this type of receiver engine, with its cranks making equal angles, is that the drop in pressure may be important.

Steam expanding from any one cylinder in series into an intermediate receiver must, in the ideal case, condense; since

$$r\frac{v_2 - v_1}{v_1} = \frac{p_1 - p_2}{p_2}.$$

Since the expansion is adiabatic,

$$\frac{v_2}{v_1} = 1 + \frac{p_1 - p_2}{p_2\gamma} = \frac{p_2(r - 1) + p_1}{p_2\gamma}$$

$$\frac{T_2}{T_1} = \frac{p_2 v_2}{p_1 v_1} = \frac{p_2}{p_1}\left(1 + \frac{p_1 - p_2}{p_2\gamma}\right) = \frac{p_2(\gamma - 1) + p_1}{p_1\gamma}.$$

Hence some condensation must occur; this value of $\frac{T_2}{T_1}$ being less than would be found for dry and saturated steam, and but little greater than for adiabatic expansion.

In the receiver engine, the less the drop of pressure at the end of the stroke, at the passage of the exhaust steam into the receiver, the less the waste. This drop is made less, cylinder-volumes being properly adjusted, as the receiver-volume is

greater and may be as entirely avoided as in the Woolf engine. This adjustment should not, however, be made at the sacrifice of net gain by expansion. The most desirable arrangement is that which, as in Mr. Corliss's compound engines, gives an adjustable expansion to each cylinder.

When proportions are adopted giving equal initial stresses, the weights of reciprocating parts, and their inertia-effects, become equalized,—a consideration of importance at very high speeds. While it is usual to seek to obtain equal work in the cylinders in series, it is probably, on the whole, generally wiser to proportion the engine for equal initial stresses; thus making all rods carry similar loads, and with equal stroke of piston, as is usual, thus securing minimum weight and cost of running parts.

9. **The Design and Proportions of a Single Cylinder** is settled by a study of the nature of the work and the cost of construction, as well as by the consideration of the problem of maximum commercial efficiency. An engine of any specified power and driven by steam at a given pressure and with a stated mean ratio of expansion, must have at any fixed speed of piston a certain easily computed volume; but it may be either of large diameter and small stroke, or of long stroke and small diameter.

An engine of short stroke will have also short rods, small crank, short frame and foundation, and less cost, as compared with a similar engine in other respects, but of long stroke of piston. Marine engines tend toward long stroke, in the merchant service, because the smaller diameter gives economy of engine-room floor-space. The short-stroke engine will have the higher speed of rotation, and a lower maximum piston-speed, and will presumably demand more careful supervision when at work, and its operation will probably involve greater actual cost, as well as risk, than the other engine with its more moderate speed of rotation and easier-worked journals. As the materials, the design, and the construction of engines are improved, the progress of change is continually in the direction of higher speeds, both of piston and of rotation; but the

prudent designer will assume a reasonably safe speed and will accept no real risk.

The *volume* of cylinder is computed as follows:

Let the work demanded be, in this cylinder,

$$U = I.\ H.\ P. = \frac{p_e LAN}{33,000};$$

in which expression p_e, L, A, and N represent the mean effective pressure, the stroke and area of piston in similar units, and the number of strokes; the units being pounds, feet, and minutes. Then the volume of the cylinder must be

$$V = AL = \frac{33,000\ I.\ H.\ P.}{p_e N};$$

in which we may vary the values A and L any way such that $AL = V = \frac{1}{4\pi} D^2 L$, when D is the diameter of cylinder.

The conditions of the market and of use have usually been found to dictate proportions ranging between

$$D = \tfrac{1}{2}L \quad \text{and} \quad D = L;$$

both in the same measure; or between

$$L = \sqrt[3]{\frac{16V}{\pi}} \quad \text{and} \quad L = \sqrt[3]{\frac{4V}{\pi}}.$$

A trifling correction being made for the area of section of the piston-rod, the size of cylinder becomes exactly fixed. The latter is a usual limiting value of L for the later "high-speed engines," the former for those of the older types. As a rule, the shorter the stroke the readier the sale, at the comparatively low price; but the longer stroke usually makes the cheaper engine in maintenance, and the waste of power in the engine itself is less.

External wastes of heat are made a minimum by adopting those proportions which give minimum area exposed in propor-

tion to weight and volume of steam used, provided all parts of the exterior waste heat equally. In this case, that cylinder which has an exterior diameter equal to its external length is best in this respect; as may be readily proved by making it a problem of maximum or minimum, thus:

Let S be the surface; then

$$V = \tfrac{1}{4}\pi D^2 L; \quad S = \tfrac{1}{2}\pi D^2 + \pi DL\cdot$$

and $$L = 4\frac{V}{\pi D^2}; \qquad S = \tfrac{1}{2}\pi D^2 + 4\frac{V}{D};$$

$$O = \frac{dS}{dD} = \pi D - 4\frac{V}{D^2};$$

and $$L = 4\frac{V}{\pi D^2} = D.$$

But, in fact, the heads and sides of the cylinder are not usually of equal heat-wasting capacity; the intermediate portions are of lower mean temperatures than the ends, and these considerations would dictate a stroke $L > D$.

The internal wastes, on the other hand, are commonly several times greater than the external, and the two heads of the cylinder, the equal areas of the piston, and the port-spaces, usually waste by far the greatest proportion of the total heat thus ejected into the exhaust. Hence, the relative heat-efficiencies of engines are the greater, for these reasons, also, as the diameter becomes smaller in proportion to length. Were all interior surfaces equally wasteful, we should have

$$V = \tfrac{1}{4}\pi D^2 L; \; S = \pi D^2 + \pi DL;$$

$$\frac{dS}{dD} = 0 = 2\pi D - 4\frac{V}{D^2};$$

$$L = \frac{4V}{\pi D^2} = 2D.$$

and the stroke should be one half the diameter of the cylinder. But in this case, also, since the heads of cylinder and piston are most effective in producing waste, they should be made relatively of less area, and we should make

$$L > 2D;$$

and in proportion to the excess of wasting capacity in the heads.

An engine of long stroke thus should excel in thermodynamic action, and, as well known, it excels somewhat, ordinarily, in dynamic, or machine, efficiency.

The quantity of steam affected per stroke varies directly as the length of stroke, other things equal, and inversely as the speed of rotation, the speed of piston being constant. The waste-fraction is, according to Fourrier, directly proportional to the square root of the time of exposure of the heat-absorbing surface; while it is inversely as the quantity of steam present. With varying speed of rotation, stroke constant, the proportion of steam so wasted thus becomes inversely proportional to $\sqrt{NL}$; with varying stroke, speed of rotation constant, this loss is inversely as the length of stroke; while with speed of piston constant, the engine will thus waste a percentage of steam supplied *directly* proportional to the square root of the number of revolutions in the unit of time. When both speed of piston and speed of rotation vary, the waste-fraction will vary as the square root of the quotient of speed of rotation divided by speed of piston, the area of piston remaining unchanged. With varying proportions of similarly-shaped cylinders, the waste-fraction will be inversely proportional to diameter or stroke, nearly. Short-stroke engines suffer more from clearance-losses and from friction-wastes than those of long stroke; while having an advantage in reduced cylinder-condensation because of their high speed, and in cost of construction.

The precise shape given the cylinder is seen in the various illustrations scattered throughout this work; each type and

style of engine having its own special design. The accompanying engraving shows a cylinder in cross-section which exhibits Mr. Porter's method of securing minimum clearance, placing a steam and an exhaust valve at each end of the cylinder, balanced carefully, and set on their edges, so that the cylinder may be drained safely.

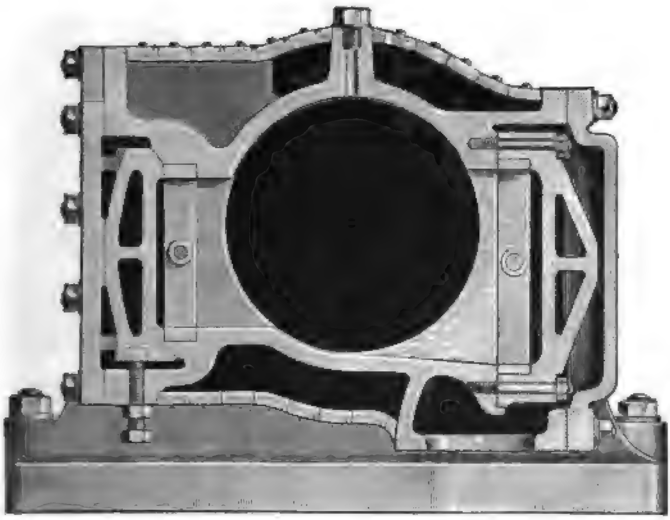

FIG. 13.—PORTER'S ENGINE-CYLINDER.

The exhaust valves are here placed opposite the steam-chest, and are in contact with only a small portion of the cylinder; so that the exhaust abstracts externally but little heat from the cylinder, and none from the entering steam.

The valves are all thus made convenient of access and capable of independent adjustment.

In Woodbury's, as in many high-speed engines, the cylinder and steam-chest (Fig. 14) form one casting. The back head is covered by a polished cap, which also covers the nuts. The front head is bolted between cylinder and frame, the stuffing-box being cast with the head. Both heads are made steam-tight to the cylinder by ground joints.

The piston is hollow and light, and has a length equal to

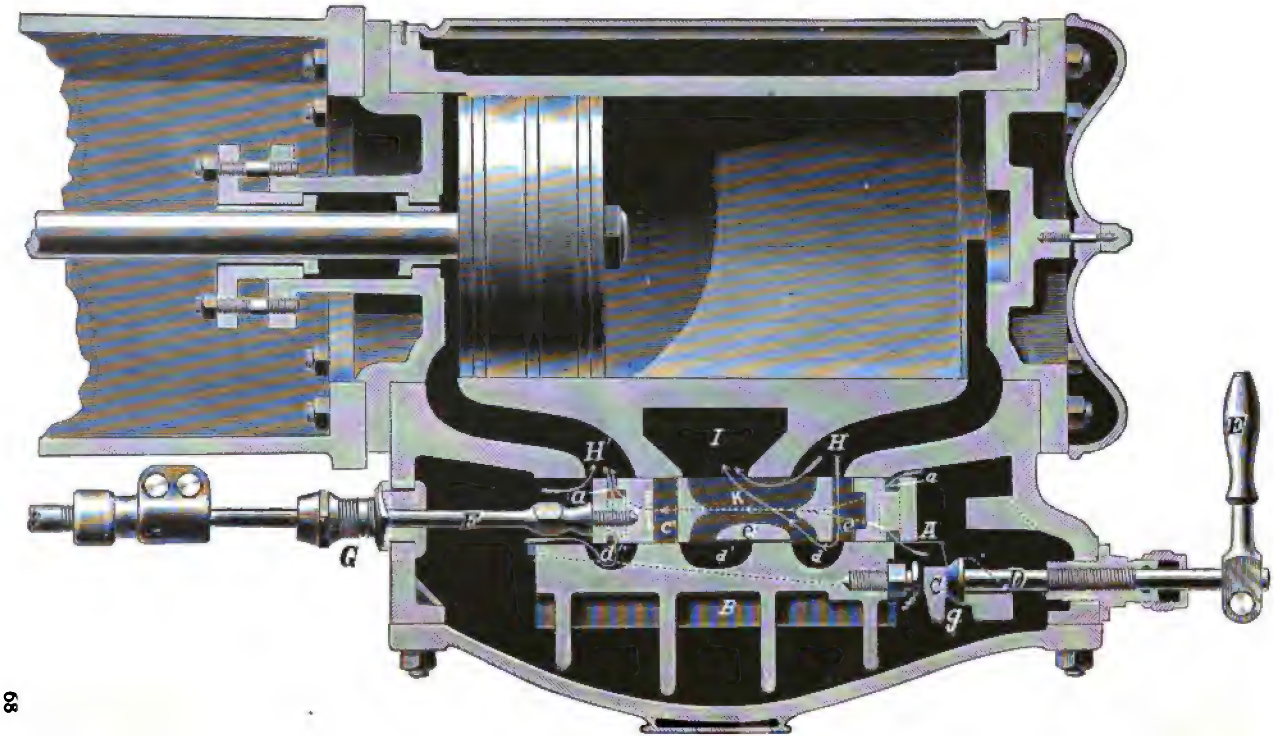

FIG. 14.—SECTION OF CYLINDER AND VALVE.

one half the diameter of cylinder. The packing consists of cast-iron rings turned eccentrically, and somewhat larger than cylinder, and cut, then sprung into corresponding grooves in the piston.

The rings do not extend all the way around, a portion of each groove at the bottom being filled with a tight section of ring. The valve A, besides taking steam at the ends, has supplemental admission-ports, a, a', which are connected by passages at top and bottom. In the position of piston shown, it has passed the centre at crank end and has moved a short distance toward the back or head end. The crank end of valve is open for admission of steam, which is entering cylinder-port H, directly past the end of valve, and also through cavity d'' in relief-plate into port a', as shown by the arrows. Steam is at the same time entering supplemental port a at opposite end at two points, and travelling through the horizontal passages into port a' and cylinder-port H'. The admission, therefore, takes place at *four points at the same time*, and, as the ports are very large, close approach to boiler-pressure is attained, and the usual loss between boiler and cylinder reduced.

A double exhaust is also used, the valve being provided with supplemental exhaust-ports, c, c'. In the position of valve shown, the exhaust steam passes from the cylinder-port H into the cylinder exhaust-passage I, in the usual way, and, in addition, is passing through supplemental ports c into central cavity K of the valve, and thence into the passage I, as indicated by arrows. This provides a very large area of opening for exit of the exhaust.

The stem D and its handle E afford a means of adjusting the packing-plate or relief-plate as it wears. The whole is a good illustration of accepted and successful practice in the construction of a flat balanced valve for use with the "automatic" system of regulation by means of a shaft-governor.

The engraving exhibits a feature of the Corliss engine of the better class, which tends to produce some economy. In this, a design by Messrs. Bullock, the exhaust-passages are separated from the main cylinder-casting in such manner as to

prevent that waste of heat by conduction from the latter to these comparatively cool passages. Both steam and exhaust chests are stayed effectively to prevent danger of yielding to pressure.

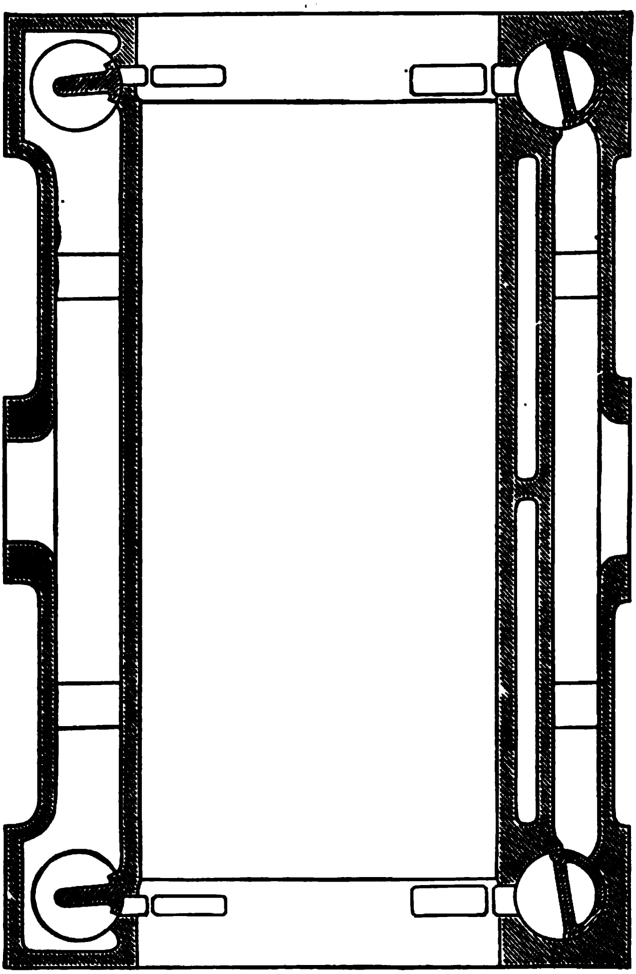

FIG. 15.—CORLISS CYLINDER. SECTION.

On the exhaust side, this becomes important, when compounding, in the small cylinder.

The Dimensions of Cylinder are first determined by the designer of a proposed engine. The data given are the steam-

pressure, the ratio of expansion, and, commonly, the speed of piston and of rotation, in feet and in revolutions per minute. The net power to be demanded is the essential, and first, datum obtained, and the "indicated" power, which governs the size of cylinder, exceeds the net, or "dynamometric," power by the amount of the internal resistances and friction of the engine. The type of engine being settled upon, the probable efficiency of the machine is known by experience, or may be approximately estimated by the designer, and the power to be calculated upon is

$$I.\,H.\,P. = D.\,H.\,P. \div E_e,$$

when E_e is the efficiency of the engine as a machine.

From the familiar expression,

$$I.\,H.\,P. = \tfrac{1}{4}\pi d^2 p_e s \div 33000,$$

the diameter of the cylinder is found to be

$$d = 205\sqrt{\frac{I.\,H.\,P.}{p_e s}};$$

when p_e and s are the mean effective pressure in pounds per square inch and the speed of piston in feet per minute.

Then, also, we have the already quoted rule,

$$I.\,H.\,P. = \frac{p_e LAN}{33{,}000},$$

when the pressure, p_e, is in pounds on the square inch, area of piston, A, is in square inches, time given in minutes, and length of stroke in feet. When the mean pressure is known, either by the process of determining the conditions for maximum duty or maximum commercial efficiency as already indicated, or by simply assuming maximum pressure and a probably best ratio of expansion, and then, calculating

$$p_e = \frac{1 + \log_e r}{r} - p_b,$$

the number of revolutions, N, to be made being given, the volume of cylinder is at once known, in cubic inches, thus;

$$\text{Vol.} = 12LA = \frac{12 \times 33{,}000 \times I.\,H.\,P.}{Np_m}.$$

In such cases, the desirable length of stroke is commonly also fixed by the designer, and the quotient,

$$\frac{\text{Vol.}}{12L} = \frac{33{,}000\ I.\,H.\,P.}{LNp_m} = A,$$

is the area of piston, in square inches and the diameter,

$$d = 2\sqrt{\frac{A}{\pi}};$$

or it may be taken out from the tables.

Thus, in marine engineering, the length of stroke is very generally fixed approximately by the size and form of the hull; the number of revolutions per minute is determined pretty closely by the speed of vessel and pitch of screw; while the steam-pressure and ratio of expansion are well settled by the practical conditions of current practice. Assume it to be required that in an engine of this kind one cylinder is to develop 500 *I. H. P.* at 75 revolutions per minute and with a stroke of piston of 4 feet; then, if the mean pressure is 40 pounds,

$$\text{Vol.} = \frac{12 \times 33{,}000 \times 500}{75 \times 40} = 66{,}000 \text{ cu. in.}$$

$$A = \frac{500 \times 33{,}000}{40 \times 4 \times 75} = 1375^{\square''},$$

$$d = 2\sqrt{\frac{A}{\pi}} = 42 \text{ inches, nearly.}$$

It is often the fact, however, that, the volume of cylinder being determined, the relations of diameter to stroke and to

speed of rotation are all fixed by determinate expressions settled by either theoretical or practical considerations, or both.

The Size of Locomotive Cylinders is usually taken to be such that the engine will just overcome the adhesion of its wheels to the rails under favorable circumstances. Mr. Webb assumes a stroke of two feet on his larger engines, and then gives them such diameter of cylinder as will give a tractive power not exceeding the adhesion under most favorable circumstances.

The adhesion of the wheel is about one third the weight when the rail is clean and sanded, but is usually assumed at 0.25.

A committee of the American Association of Master Mechanics, after studying the performance reports of the best engines, proposes the following formula for weight on driving-wheels:

$$W = \frac{0.85\, Cd^2 PS}{D};$$

in which the mean pressure in the cylinder is taken as 0.85, the boiler-pressure at starting; C is a numerical coefficient of adhesion, d the diameter of cylinder in inches, D that of the drivers in inches, P the pressure in the boiler in pounds per square inch, S the stroke of piston in inches. C is taken as 0.25 for passenger engines, 0.24 for freight, and 0.22 for "switching" engines.*

Where fuel is costly, larger cylinders may be employed to give greater expansion, and compounding and steam-jacketing may prove, on the whole, economical. The common forms of valve-gearing, however, are not adapted to a higher ratio of expansion than 4.

The common builders' rule for determining the size of cylinders for the locomotive is the following, in which we accept Mr. Forney's assumption that the steam-pressure at the engine

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* Railroad and Engineering Journal; Sept. 1887.

may be taken as nine tenths that in the boiler.* The tractive force is, approximately,

$$F = \frac{0.9p_1 \times A \times 4s}{C};$$

where C is the circumference of tires of driving-wheels, s the stroke in inches. Taking the adhesion at one fourth the weight, W,

$$F = 0.25 W = \frac{0.9p_1 \times A \times 4s}{C};$$

whence the area of each piston is

$$A = \frac{0.25CW}{0.9 \times 4 \times p_1 s}.$$

The thickness of cylinder is determined, in small engines and at low working-pressures, by quite other considerations than those of strength to resist internal pressure. It must be made of a hard, strong iron, to resist wear, and is therefore comparatively lacking in ductility; as it must be cast and cooled in the foundry, it must not have abrupt variations of thickness in its various parts, or any sharp angles; it must be thick enough to bear reboring; and it must be, above all things, *stiff* enough to keep its shape under all circumstances. It must, therefore be made comparatively heavy, and much thicker than is ordinarily demanded to safely carry the proposed steam-pressures.

Very large cylinders and at high pressures, computed with a good factor of safety, on the other hand, if strong enough, will be usually stiff enough if they are set vertically. Horizontal cylinders, if large, rarely retain their form perfectly, and not infrequently crack after a short life. Experience only can settle the best proportions.

Where one head is cast in, and especially with short cylin-

* Catechism, Interrog. 708.

ders such as are common in marine practice, greater stiffness and strength are secured.

For engines of the older types and under moderate steam-pressures, some builders have for many years restricted the stress to about 2550 pounds on the square inch, and

$$t = ap_1d + b$$

is a common proportion; t, d, and b being thickness, diameter, and a constant added quantity varying from 0 to $\frac{1}{2}$ inch, all in inches; p_1 is the initial unbalanced steam-pressure. In this expression b is made larger for horizontal than for vertical cylinders, as, for example, in large engines 0.5 in the one case and 0.2 in the other, the one requiring reboring more than the other. The constant a is from 0.0004 to 0.0005: the first value for vertical cylinders, or short strokes; the second for horizontal engines, or for long strokes.

Cylinder-heads are to be made strong enough to bear working pressures, and with a moderate factor of safety, as 6. In considering rectangular plane surfaces subjected to fluid pressure, Weisbach deduces the following formula for square plane surfaces, which will answer for either cylinder-heads or steam-chest covers:

Let t_1 = the thickness in inches of the cylinder-heads;
P = the boiler-pressure in pounds per square inch;
d = the diameter of the steam-cylinder in inches.

$$t_1 = 0.003\, d \sqrt{P}.$$

With large cylinders the heads are stiffened by radial ribs.

Cylinder-heads may be given a thickness, at the edges and in the flanges, exceeding somewhat that of the cylinder. An excess of not less than 10 nor more than 25 per cent is usual. It may be thinner in the middle. Where made, as is usual in large engines, of two disks with intermediate radiating, connecting ribs, or webs, that section which is safe against shearing is probably ample. A shearing-stress equal to that taken as safe in tension, in computing thickness of cylinder, is allow-

able. An examination of the designs of experienced builders, by the Author, gave the following,

$$t'' = \frac{Dp}{3000} + \tfrac{1}{4}'',$$

as the usual thickness, in inches, D being the diameter of that circle on which the thickness is taken. This gives a factor of safety of about 7.

Cylinder-flanges should be a little thicker than the cylinder, and are usually made of equal thickness with those of the heads. They should be made only just wide enough to take the bolts. A common proportion is one fourth thicker than the heads at their edges inside their flanges.

The cylinder-heads are sometimes a single disk, sometimes double; the space between being braced by radial connecting webs. The thickness of the head is usually made about the same as that of the cylinder, but it should vary as the diameter and as the square root of the pressure,* and a rule giving good results is expressed by

$$t = 0.005\, d \sqrt{p_1} + 0.25,$$

when the dimensions are in inches and pressure in pounds on the square inch. Thus, for the cylinder above calculated, when $d = 200$ and $p = 125$,

$$t = 0.005 \times 20 \sqrt{125} + \tfrac{1}{4} = 1.37 = 1\tfrac{3}{8},'' \text{ nearly.}$$

As the flanges of the cylinder-heads are commonly a trifle thicker than the head, the angles at which the flanges join the main casting should be rounded, "filleted," rather than left sharp, to avoid danger of cracking.

When the thickness computed exceeds $1\frac{1}{4}$ or $1\frac{1}{2}$ inches it is advisable to make the head of two disks united by radial webs.

The Cylinder-head Bolts may be either small in number and large in size, or the reverse; but they should always be so closely spaced that no springing of the flanges of the head, and

* Weisbach; vol. II. sec. ii. § 412.

consequent leakage, can occur. This spacing should probably not often exceed four or five times the thickness of the flanges. For example, a dozen bolts on a twenty-inch cylinder is a good number, the thickness of flanges being $1\frac{3}{8}$ inches. They should be of such size as not to be subject to liability of being twisted off by a careless workman, and not to be strained by the steam-pressure. A maximum stress of 5000 to 6000 pounds per square inch is considered a good figure. Bolts one inch in diameter would thus be ample in the case above assumed.

Flat surfaces and members are usually designed as seldom as possible, and made of as small area as practicable. They are commonly given very nearly the thickness of the heads, and, if the steam-pressure is very great, are strengthened by ribs arranged to suit the design or the ideas of the designer. It is desirable that the thickness of the whole casting be made as uniform as possible to avoid danger of shrinkage stresses, strains, and cracks.

Relief-valves should be fitted to all engines.

Figure 16 shows an excellent form of automatic relief-valve. The engine is fitted with two of these in place of cylinder-cocks. They can be set to open at any desired pressure and afford a direct and sure relief. This avoids any of the disastrous effects of water in the cylinder. They can also be operated by hand as ordinary cylinder-cocks.

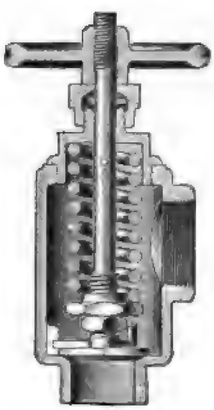

FIG. 16.—RELIEF-VALVE.

As here seen, they are held to their seats by a spring adjusted to a safe pressure in excess of the maximum working pressure, and will thus lift automatically before the pressure becomes dangerous.

Jacketed and unjacketed cylinders are proportioned, usually, substantially alike. Since, however, the jacketed engine is less subject to internal heat-wastes, and since its wastes are more nearly uniform throughout, both outside and in, the considerations already stated would indicate the advisability of adopting, on the

whole, a comparatively short stroke. In other words, jacketing is more desirable on short- than on long-stroke engines. In practice, this, which is perhaps an hitherto unobserved fact, has not modified proportions in any respect.

The jacketed cylinder commonly consists of two shells, the cylinder proper, and the "liner," which latter is now commonly of rather soft steel. The former is proportioned in accordance with the principles and rules already given. The liner is treated as a separate cylinder liable to either crushing by the jacket-steam or bursting by the pressure driving the piston. It is commonly thus given a thickness less by 15 to 20 per cent than that of the outer shell. A rule adopted by British builders makes it

$$t'' = \frac{p_1 D}{2500} + 0.02.$$

Where both parts are cast in one, it is common and advisable to make the outer cylinder with some provision for permitting expansion and contraction to produce changes of form without strain, as is illustrated in the designs of Corliss and of Leavitt.

A steel liner of proper composition and initially sound will probably always prove permanently reliable. Whitworth steel liners, for example, used for many years in the British navy for this purpose, have never been known to crack. If rightly made, they need not exceed ½ inch in thickness for even large engines.*

10. Steam-pipes, and Ports, and Valves should be so proportioned as not to appreciably reduce the pressure of steam by "throttling" or "wire-drawing" it on its way from boiler to engine. The custom with designers has commonly been to make the area of cross-section of steam-pipe, and the clear area through its valves and the steam-ports, a specified

* London "Engineer"; June 17, 1887, p. 484.

fraction of the area of the cylinder, as from one eighth to one sixteenth.

With short pipes, the area of section and the diameter adopted may be easily and satisfactorily settled by assuming a rate of flow of steam through them which shall not exceed a specified figure, as, for example, 100 feet (30 m.) per second or 6000 feet (183° m.) per minute. In such case, the area of steam-pipe and port, for example, would be not less than

$$a = \frac{ALN}{6000}$$

where measured in square feet, and the area, length of stroke, and number of strokes are taken in feet and minutes. The ratio

$$\frac{a}{A} = \frac{V}{6000}$$

is thus that of speed of piston, V, to the assumed maximum rate of flow.

Exhaust ports and pipes must be made considerably larger, usually, than the steam ports and pipes; since they must discharge all the steam which enters the cylinder in the period of admission in the comparatively brief time, the small fraction of that period allowed at the end of stroke, while the engine is "turning the centre." An excess of 25, or even 50 per cent, is considered by designing engineers none too much.

Mr. Babcock, who has collected recent work on the subject, obtains the following table for the approximate weight of steam per minute which will flow, with a loss of pressure, due to friction, of one pound per square inch, through different sizes of straight pipes, having a length of 240 diameters:

For horse-power, he would multiply the figures in the table by 2. For any other loss of pressure, multiply by the square root of the proposed loss. For any other length of pipe, divide 240 by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient, which will give the flow for 1 pound loss of pressure.

FLOW OF STEAM THROUGH PIPES.

Initial pressure by gauge, lbs. per sq. in.	Diameter of Pipe in Inches. Length of each = 240 diameters.													
	$\frac{3}{4}$	1	$1\frac{1}{4}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	15	18
	Weight of steam per minute in pounds, with 1 pound loss of pressure.													
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4	804	1177
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	996	1458
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5	824.1	1318	1930
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0	1450	2122
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1567	2294
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1009.2	1486.5	2378	3481

The quantity of steam flowing through a pipe under a given "head" or loss of pressure, increases directly as the square root of the density, and of the loss of pressure, and inversely as the square root of the length.

The figures just given were computed by the formula

$$W = 303.25\, d^2 \sqrt{\frac{D(p_1 - p_2)}{L\left(1 + \frac{3.6}{d}\right)}};$$

in which d = diameter in inches, D = density or weight per cubic foot; p_1 = the initial pressure, p_2 = the pressure at end of pipe, and L = the length expressed in diameters. For sizes of pipe below 6-inch, the flow is calculated from the actual areas of "standard" pipe of such nominal diameters.

The resistance to the steam entering the pipe is equal to the friction of 60 diameters additional length. A globe valve may be estimated as 60, and each elbow as 40 diameters length. These equivalents must be added in all cases in estimating the length of pipe.

Where the steam escapes into a pressure less than half that in the boiler,

$$HP = \frac{(p + 15) \times 12ak}{7};$$

where p = pressure of steam by gauge, a = area of orifice in square inches, and k is a coefficient = .93 for a short cylindrical pipe, or .63 for a thin opening, or for a safety-valve.

Where the steam flows into a pressure more than half the pressure in the boiler:

$$HP = 3.8\, ak \sqrt{(p + 15 - d)\,d};$$

in which d = difference in pressure between the two sides, in pounds per square inch, and a, p, and k as above.

Where a given horse-power is required to flow through a given opening, to determine the necessary difference in pressure:

$$d = \frac{p + 15}{2} - \sqrt{\frac{(p + 15)^2}{4} - \frac{HP}{14\, a^2 k}}.$$

To reduce either of the above to pounds per hour, multiply HP by 30. For pounds per minute, divide HP by 2.

The steam-pipes of all engines should be so arranged as to insure effective separation and return to the boiler of all water primed over with it or produced by condensation in the pipe *en route*. This is usually made certain, at least to the extent of avoiding danger from water entering the engine, by the introduction of a "separator" near the engine, as in Fig. 17, which exhibits a system of piping for a "high-speed" engine, with its separator, drain-pipes, and steam-trap attached, as arranged by Mr. Stratton. As here shown, the separator is of larger size than is customary, but, with fast-running engines and a hard-worked boiler, not too large for safety.

The steam-trap, or its later equivalent, the "steam-loop," automatically returning the water to the boiler, is an essential feature of the system.

The "*steam-loop*" is an arrangement of return-water piping which permits the water caught at the separator to flow back without the interposition of the steam-trap, as seen in the illustration. It is a siphon, in which the flow is produced by a difference of head, in the two legs, produced, not by difference

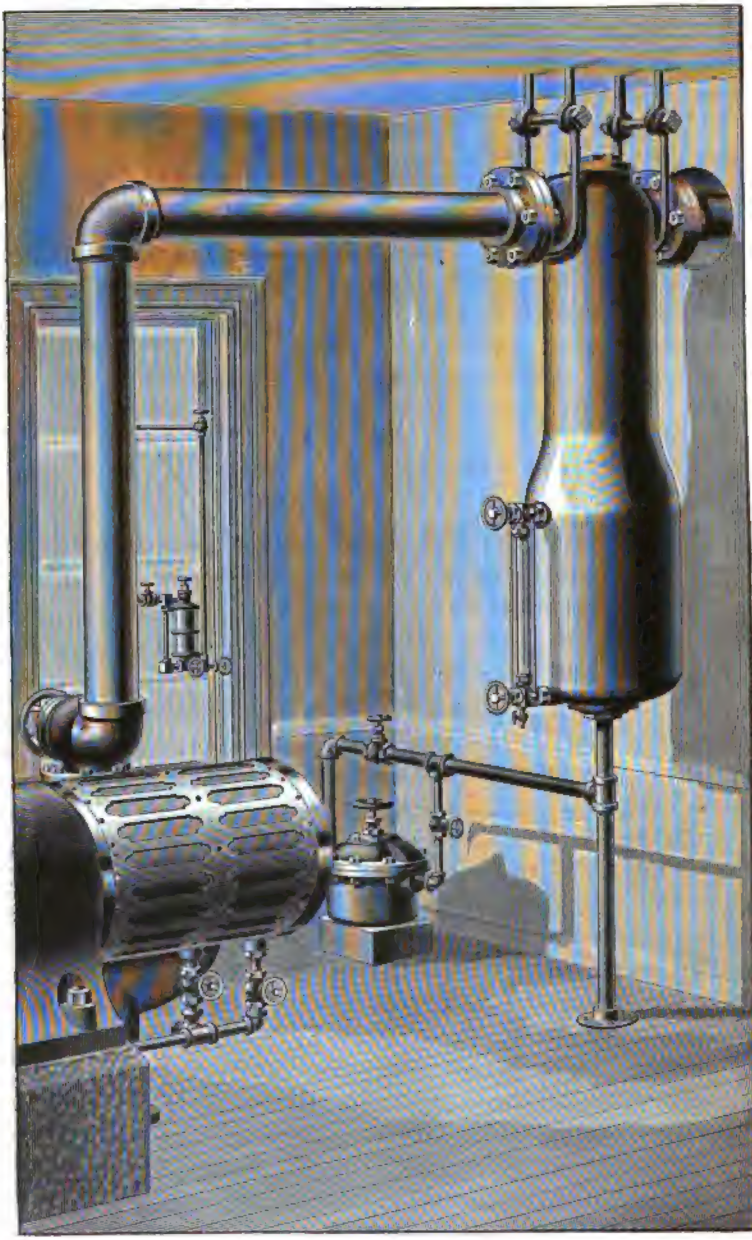

FIG. 17.—STEAM-PIPE, SEPARATOR, AND TRAP.

of length, but by difference of density; that leading from the engine into the horizontal pipe above being filled with a mixture of steam and water, which settles into solid water in the drop-leg at the boiler. The water is thus returned continuously and at nearly full boiler-temperature.

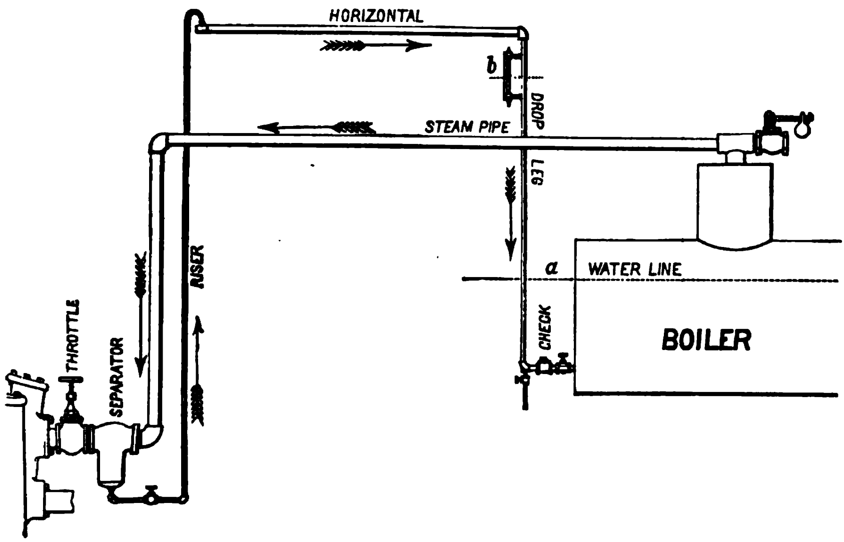

FIG. 18.—THE STEAM-LOOP.

The quantity of water which may be separated by such methods as are here described is determined by the proportions and construction of the separator. Its diameter should not be less than twice that of the steam-pipe, and its depth should be sufficient to receive water occasionally suddenly thrown over by the boiler with the steam; this depth should not be less than three or four times its own diameter and is often made greater. A good separator should keep the priming under 1 per cent the weight of the steam, if originally of a quality as high as 95 per cent. It is always important that the separator should contain not only a reservoir of considerable volume, but it should be somewhat isolated by a diaphragm or other equivalent device.

Exhaust steam-passages should not be permitted to traverse the cylinder in contact with the walls of steam-spaces, whether of the steam-ports or of the cylinder itself; as such a construc-

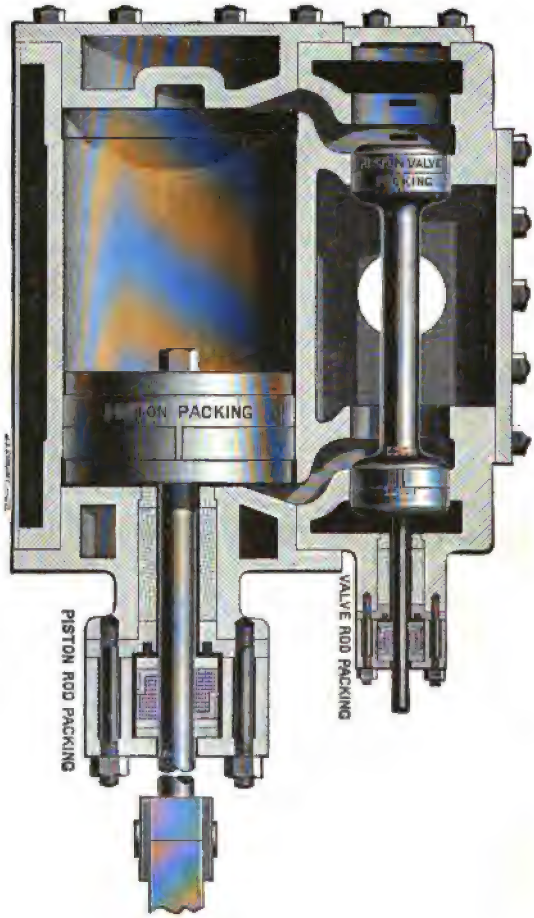

FIG. 19.—METALLIC PACKINGS.

tion causes loss of heat by direct transfer. This point is generally attended to carefully by good designers.

The disposition of metallic packing is illustrated in the figure; which shows that used for valves and for piston, the

usual ring, and the form of metallic packing often employed in stuffing-boxes of piston and valve-rods. That here shown is a form invented by Mr. Tripp. Tuck's packing is composed of alternate layers of canvas and rubber; Selden's is of suitably prepared paper; Martin's has a bearing layer of brass or other wire gauze. Many forms of solid metallic packing are used on large engines.

The use of metallic packing is only admissible with well-constructed work; as it usually cannot accommodate itself well to rods out of line or of varying size. Where good workmanship can be depended on, a solid sleeve of white-metal on the piston-rod or valve-stem as made by Professor Sweet, or even a simple fit in a long solid bearing, as in Mr. Harris's engines, may be resorted to with satisfaction.

11. **Pistons** are, as "running parts," subject to heavy static and kinetic stresses through the action not only of steam-pressure, but also of the jar inseparable to some extent from motion, in all forms of machinery, and of the shocks due to the introduction of water into the cylinder and other less dangerous accidents.

The piston is liable to three principal methods of destruction: fracture through stresses primarily coming of ordinary steam-pressure and resisted by its rod; similar strains due to water in the cylinder, filling the "dead-space," and resisting its approach to the head; abnormal strains produced by the pressure of its edge and its rings against the surfaces of the cylinders and causing "cutting" and consequent strain. It is easy to provide with certainty against the first of these stresses, as it is perfectly determinable; but the others are the usual cause of accident to this member, and are quite impossible of calculation. General experience, rather than determined principles of calculation, has therefore settled its proportions.

The piston is commonly—and in large engines invariably—made of a pair of disks separated by a space varying from one half, or even its full, diameter, in small engines, to as little as one tenth or one fifteenth in very large engines. These disks are sometimes plane and parallel, sometimes concave toward

each other, and often of forms determined by the configuration of the cylinder-heads and their stuffing-boxes. Occasionally a single disk is employed; in which case the surface is usually more or less curved. When fitted with rings having considerable breadth of surface, the latter bear on the cylinder, and, in the horizontal engine, may take the wear. With other arrangements, in which the ring is simply a packing device, the piston itself carries all the weight.

12. **Piston-rods** may be treated as columns subjected to compression upon rounded ends; although often taken as having one end fixed, the other rounded. The length, from its junction with the crosshead to its insertion in the piston, varies from 1.1 the piston-stroke in beam-engines of long stroke, to 1.5 in high-speed and short-stroke marine engines. Its length ranges from 8 diameters in very short strokes, to 10 or 12 diameters in the longer connected engines, and to twice the latter figures in special cases.

For short and heavy rods it will be quite sufficient to proportion their section to carry not to exceed one half the stress corresponding to the elastic limit. Long rods may be designed by the Gordon formula for columns,

$$p_1 A = \frac{fa}{1 + b\frac{16}{9}\frac{l^2}{d^2}};$$

in which the total maximum load, $p_1 A$, is given as a function of a, the sectional area of the rod, and $\frac{l}{d}$, the ratio of length to diameter; and in which the factor of safety, f, is to be not less than 6 and is commonly considered to be better if made 8; while the constant, b, is 0.003. The least section, in tension, should be made not less than

$$a' = \frac{p_1 A}{8000}.$$

Professor Unwin finds the expression,

$$d'' = bD\sqrt{p_1};$$

in which D is the cylinder-diameter, and the constant, $b = 0.0167$ for iron, and $b = 0.0144$ for steel. An examination of a considerable number of rods in use gives

$$d'' = \sqrt[4]{\frac{D^2 p_1 l^2}{a}} + \frac{D}{80}, \text{ nearly};$$

in which the constant, a, $= 10,000$ and upward in the various types of engine, the marine direct-acting screw-engine or ordinary fast engines on shore giving the lowest values; while "low-speed engines," being less liable to accident from shock, give $a = 15,000$, often.

Where rods are fitted by shrinking into their sockets, they should be given the diameter

$$d_1'' = 1.0025 d_2;$$

when d_2 is the bored diameter of the socket. The firmness of the connection is tested by sounding with a hammer. In small engines, no other fastening is needed, if the shrinking be properly done.

When secured by a tapered end and a key, the taper should be not less than 1 to 16 in cases where the thrust is taken by a collar at the end of the coned part, and not more than 1 to 8 when held by properly proportioned nuts. One end must be so constructed that it may be passed through the stuffing-box in assembling. If threaded, the shearing-stress should not exceed one fourth the elastic limit of the material—about 5000 for iron, and 7000 to 9000 for steel; which limits are good figures for those parts in tension. $D =$

For a first case assume the diameter of cylinder, $\cancel{d} = 60$ inches; the stroke, $L = 6$ feet; and the pressure, maximum unbalanced, $p_1 = 60$ pounds per square inch. Then

$$d'' = \sqrt[4]{\frac{60^2 \times 36 \times 60}{6000}} + \frac{1}{2} = 6.5,$$

If this were an engine of high speed of rotation, as, for example, a marine screw-engine,

$$d'' = \sqrt[4]{\frac{60^2 \times 36 \times 60}{4000}} + \frac{1}{2} = 1.14 = 7\tfrac{1}{8}'', \text{ nearly.}$$

For a screw or other high-speed engine, take $d = 20$; $L = 2$; $p_1 = 125$; then

$$d'' = \sqrt[4]{\frac{20^2 \times 4 \times 125}{4000}} + \frac{1}{2} = 3.15 = 3\tfrac{1}{8} \text{ inches nearly.}$$

Were this an engine of moderate speed, the figures would become

$$d = \sqrt[4]{\frac{20^2 \times 4 \times 125}{6000}} + \frac{1}{2} = 2\tfrac{7}{8}, \text{ nearly.}$$

In these formulas, the length of stroke, rather than of rod, is taken, since the unsupported portion is most nearly measured by that length.

In any case in which the rod may be considered a short column, as when less than twelve or fifteen diameters long, the section may be made a constant fraction of the product of area of piston and steam-pressure, and allowed 5000 pounds per square inch in either tension or compression, the section being in each case measured at the point of minimum area and maximum stress.

Steel rods are commonly made of a grade approaching wrought iron as nearly as may be in its characteristics. Hard steels, such as are used for tools, and even machinery steel, so called, are seldom used. Mr. Metcalf reports that experience shows good crucible steel containing 0.8 per cent carbon to be superior to either harder or softer metal for the rods of steam hammers, probably the most trying of all known cases of such use. Such steel has a tenacity of about 100,000 pounds per square inch (7030 kgs. per sq. cm.); but softer grades are more

generally employed, and two thirds the above figures may be assumed in computations for designs. The size of rod may be reduced by correspondingly increasing the divisors in the formulas, and using

$$t'' = \sqrt[4]{\frac{D^2L^2p_1}{10,000}} + \frac{1}{2},$$

$$t'' = \sqrt[4]{\frac{D^2L^2p_1}{6500}} + \frac{1}{2},$$

for the two cases taken.

Where connecting the rod to the piston at the one end, and to the cross-head at the other, it must be so made as to be safe against the pull of the rod; the thrust may be safely allowed to take care of itself at those points. The methods of connection are very various in different engines and as planned by individual designers. In some cases the rod is screwed into the socket at either end and secured either by a key, a pin, or a lock-nut. In other designs the rod is enlarged at the piston end, turned to a tight fit, and forced into the hub of the piston by the heavy pressure of the hydraulic press or a powerful "jack," and the projecting end or shoulder riveted over as an additional security. In still other instances the rod is nicely fitted and held in place by a heavy key, driven through hub, or the cross-head, as the case may be, and the centre line of the rod. An old design gives the rod a taper and fits it into a correspondingly tapered hole in the piston or cross-head, and holds it in place by a heavy nut. This is perhaps the best plan, for very large engines, at the piston end, and the key may be used at the cross-head end. The last-named device is the most common in general practice. The Author has used the screwed end on small engines with satisfaction; in marine work he has been accustomed to see the tapered end and key principally used.

The precaution to be observed in proportioning these con-

nections is to see that the sections subjected to pull or to shear have a factor of safety of at least eight or ten, and that they can be easily and cheaply constructed. Gibs or "cotters" should be made so as not to seriously weaken the parts into which they are inserted.

Guiding mechanisms for the head of the piston-rod should be proportioned, not only to safely carry the load to which it may be subjected, but also in such manner as to insure free and unintermitted lubrication and safety against heating at maximum pressure and velocity of rubbing.

Parallel motions, such as have been described in Chapter III, Part I, consisting of a linkwork with pin-connections, are proportioned as struts and their pin surfaces as journals, according to the rules to be given for connecting-rods and for other journals. They should be arranged in such manner as not to be liable to being thrown out of line by maladjustment, and it is ordinarily considered best to make them without provision for taking up by strap-ends and keys; but to give each end a solid eye. Wear, with ordinarily good usage, will not be likely to give trouble; but the pins are sometimes coned to permit taking up slight wear by a lateral adjustment.

Cross-heads with guides are the most common devices, at present, for steadying the head of the piston-rod against the lateral stress of the connecting-rod. The cross-head is usually fitted with broad "shoes" or "gibs" to take the wear on the guides; the pin is commonly secured in the cross-head, the connecting-rod end being fitted with "brasses" to serve as bearings; but an equally good—and, under some circumstances, probably, even a better—arrangement is the reverse; the pin is fast in the rod-end and the bearing is a part of the cross-head. Fig. 21 shows the former arrangement, and exhibits a device for adjusting the shoes to take up wear.

This, which is the intermediate element of the train of mechanism between piston-rod and connecting-rod, is subject to the same stresses, modified, however, by the introduction of some lateral components.

The Cross-head is the piece which unites the two rods and

holds their point of junction in line against the deviating stresses coming of the continually changing angularity of the connecting-rod. It thus must be proportioned to meet a variety of efforts, as well as given the right shape to glide along the guides with minimum friction and heating under lateral pressure. It must also properly carry the cross-head pin, to which the connecting-rod end is attached, safely and rigidly.

In some cases, as in some locomotives, the cross-head consists of simply a solid block, of nearly rectangular section, fitted to take the head of the piston-rod at its end, and carrying a "wrist-pin" in a slot of such width as may be needed to receive the connecting-rod end. In stationary engines, and in beam-engines, the cross-head consists, often, of a heavy block taking the head of the piston-rod, and of attached lateral arms carrying, at their ends, the "gibs" which engage the guides. The cross-head is connected to the end of the beam by links, or short connecting-rods, each having a length about equal to one half the stroke of piston. The sizes of these arms are to be computed as of beams fixed at one end and loaded at the other.

The pressure on the cross-head guide is the lateral component of the thrust of the connecting-rod, and is measured at any moment by the tangent of the angle of deviation of the rod from the centre-line of the engine. This angle and the pressure, when "following" full stroke, are maxima when the crank and rod are at right angles, and the ratio of lateral pressure to the thrust on the rod is then tan ϕ and equal to the ratio of the length of crank to that of rod.

The broad "slipper-guide" illustrated in some marine engines and in a few stationary engines is an excellent form where it is not anticipated that the engine will work backward *either* frequently or at full power. For locomotives, both working surfaces must have ample area.

The rubbing surfaces are so proportioned in what is generally considered good practice that, if V be their relative ve-

locity in feet per minute, and p be the intensity of pressure in pounds per square inch,

$$pV < 60{,}000$$

and

$$pV > 40{,}000.$$

The lower is, of course, the safer limit; but for marine and stationary engines it is allowable to take

$$p = \frac{60{,}000}{V},$$

and the Author has found this a successful proportion in his own experience and practice; or, according to Rankine, for locomotives,

$$p = \frac{44{,}800}{V + 20};$$

where p is the pressure in pounds per square inch, and V the velocity of rubbing in feet per minute.* This includes the sum of all pressures forcing the two rubbing surfaces together.

Some British builders of portable engines restrict the pressure between the guides and cross-head to less than 40, sometimes 35, pounds per square inch. The total pressure is then

$$P = pa = \frac{Ap_m r}{l};$$

where p, p_m, a, and A are the pressure as above, the mean effective steam-pressure, the area of cross-head surface and of engine-piston, respectively.

The work lost in friction is

$$U_f = T \tan \phi V.$$

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* Friction and Lost Work in Machinery; R. H. Thurston; N. Y., J. Wiley & Sons, 1885; p. 241.

The area of contact and rubbing should usually exceed

$$a = 0.025\,U \text{ for iron;}$$

$$a = 0.017\,U \text{ for steel.}$$

It should never be so small as to allow observable wear or heating. It may sometimes be found impracticable to secure as large an area of rubbing surfaces as are computed by the methods given above; and, in such cases, the provisions for insuring effective, and especially continuous, lubrication must be very carefully attended to.

The cross-head, as employed on stationary engines, is given many forms and an equal variety of proportions. That here illustrated is Mr. Porter's form. It is made a solid block of

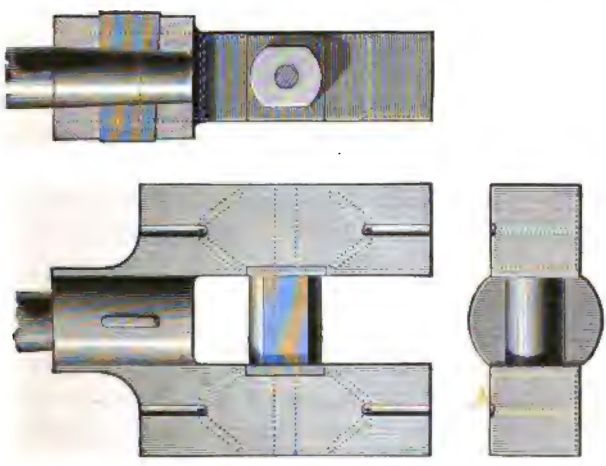

FIG. 20.—THE PORTER CROSS-HEAD.

cast-steel with its pin in the middle of its length. Its surfaces and those of the guide-bars are finished by scraping to true planes. Being, in this form, incapable of deflection, it is equally in contact with the guides at all points; and the thrust from the angular vibration of the connecting-rod is equally distributed over its surface.

The pin is made separately, of steel, flattened on top and bottom, thus insuring that the boxes can never bind on the pin at the extremes of the vibrations of the rod—a point of some importance.

The form of cross-head next illustrated is designed for small engines, usually with vertical frames, and is a simple and satisfactory construction. The guides are *bored* to meet the

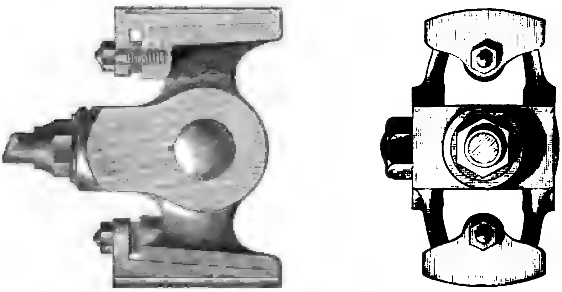

FIG. 21.—SMALL CROSS-HEAD.

adjacent parts of the cross-head ; and the latter are reinforced with adjustable bearing-pieces, turned to fit, and set out by a longitudinal sliding motion on suitably constructed inclined planes. The pin is opposite the middle of the bearing, and the rod is thus freed from a liability to spring,—observed where the pin overhangs.

Several forms of large cross-head are shown on the various illustrations of larger engines.

The cross-head pin, or wrist-pin, is generally made of generous proportions, for the reasons that the relative motion of its bearing and journal is such as to produce a tendency both to wear oval in section and to rub out the lubricant from between their surfaces. The pin is usually secured in the cross-head and movable in the head of the rod; but it is sometimes made fast in the rod, and the rubbing surfaces extended by division between two parts, vibrating within the cross-head. By this arrangement it is possible, in many cases, to get a more extended surface and greater certainty of lubrication. The total length of journal should, if possible, be made at least suf-

ficient to reduce the pressure well below the minimum given by the preceding formulas as proposed by Rankine or by the Author.

14. **The Connecting-rod** constitutes one of the most important details of the engine, and its design involves the study of a greater variety of conditions of operation, of stresses, and of construction in detail than any other element of the machine. It must bear not only the pull and the push of the whole steam-pressure on the piston; but it is also subject to the side-strain of its own often immensely rapid lateral surges at each half-revolution of the crank. It must receive and deliver the transmitted energy across surfaces of necessarily small area, and, in the case of the crank-pin, of high velocity of rubbing. It must itself be as small in section and as light as is consistent with safety, and yet must be provided with convenient and safe means of alteration of its length to take up the wear of its "brasses." The forms of connecting-rod in use have been described elsewhere.

The material of the rod is often forged iron, but is now most usually, in the best construction, forged steel. Small engines are often fitted with cast rods of machinery or soft steel, or, in same cases, of malleableized cast-iron.

Forged-steel connecting-rods are advisable, particularly on high-speed engines, on account of superior strength and reduced weight. The connecting-rod (Fig. 22), as designed by the builders of such engines, has solid ends, milled out to receive the brasses, which are held in place by a cap on the crank-pin end and by the cross-head on the other.

The brasses are set up by wedges, the full width of the connecting-rod, drawn down by steel bolts until the brasses are forced together solid. The upper bolts are used to lock the wedges. There are no gibs or keys in this design to work loose, and the wedges will not strain the strap.

Often the rods employed in the smaller sizes of engine are steel castings, as shown in Fig. 23, and those of the larger sizes of forged steel, with "marine" style of box on the crank end. The rods are secured to the cross-head pin with feathers

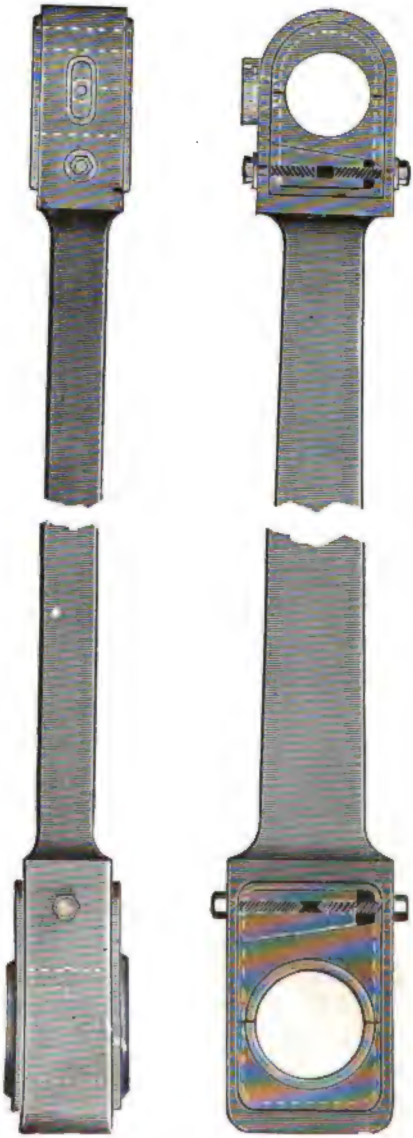

FIG. 22.—STEEL CONNECTING-ROD.

and binding-bolts, and the crank-boxes are hammered babbitt metal, bored and scraped to fit.

Test-pins of considerable length, ground to perfect cylinders,

are often used in each end, and the holes tested for parallelism and truth, and scraped until perfect accuracy is secured. The crank-caps are secured by studs riveted in, and safety-nuts used for adjusting the cap.

The illustration, herewith given, exhibits the standard form of the rod designed by Mr. Porter.

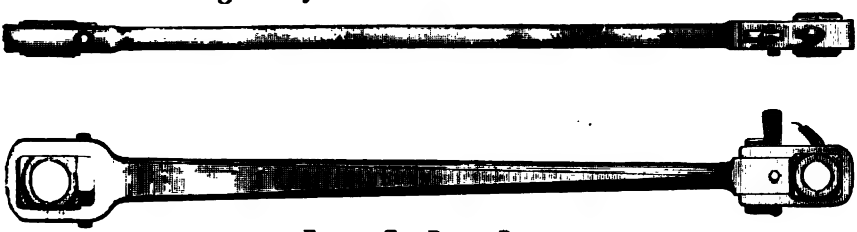

FIG. 23.--THE PORTER ROD.

It is made with the ordinary strap at the cross-head end, but the crank end is solid. This is a valuable feature. Under rapid vibration it is absolutely safe. There is no strap liable to spread, nor key liable to be thrown out.

The rod is deepened by a uniform taper towards the crank end, where the depth is in excess of that required for rigidity under any strain from rapid vibration.

The accompanying figures show a usual form of rod-end for marine work and for fast stationary engines. In this design, by Bullock, there are no narrow keys, but all adjustments

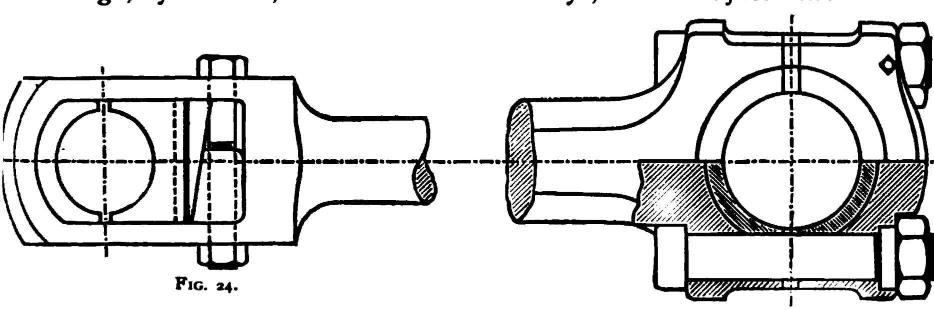

FIG. 24.

CONNECTING-ROD ENDS.

FIG. 25.

have full bearings, and the brasses are given solid bearings the full width of the rod. In the main or crank-pin end (Fig. 25) the adjustment is made by two steel bolts fitting reamed holes. This

is probably the smallest, lightest, and strongest form of head for large pins. The brasses are turned to fit bored holes, thus securing uniform and perfect bearing.

The cross-head end (Fig. 24) is adjusted by a wedge and bolt, giving a large bearing on the brass. All "shimming," after the wedge has made its full travel, can be put on straight faces, shown by dotted line in Fig. 24, instead of on the wedge face. The rod should be forged in one piece in all sizes.

The succeeding figure is a reduced working drawing of a standard rod of the preceding type, and exhibits every dimension drawn to scale.

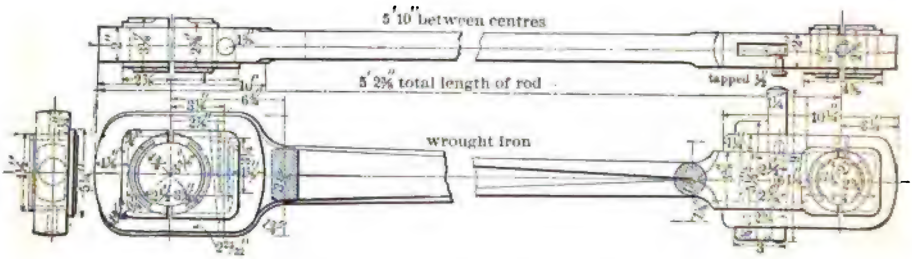

FIG. 26.—WORKING DRAWING OF ROD.

FIG. 27.—CONNECTING-ROD.

For the smaller sizes and single-acting engines, the accompanying design, the Westinghouse rod for compound engines, is a very excellent form, giving a convenient arrangement for taking up, a strong rod in compression, and very secure fastenings.

The rod consists of a body and two boxes, with a take-up wedge between the body and the top of the lower box. Both the upper and lower boxes are in halves, made of hard brass and babbitted. A continuous strap has its ends bolted solid to the lower half box, and runs around the upper box, binding the whole strap together. It is obvious that tightening up on the wedge-bolt will expand the rod lengthwise, and take up both ends alike and to any desired adjustment. The strap transmits no strains, except whatever unbalanced upward pull may

remain on the low-pressure piston when running under vacuum. All parts of the rod are of hammered steel, and the fits are milled to gauge.

The Woodbury system of construction of connecting-rod will be readily understood by an inspection of Fig. 28. The body of rod, *A*, is of **I** section, securing the maximum stiffness with minimum weight, and it tapers, as shown, from cross-head to crank-pin end. The strap, *B*, is secured in position by the key-bolt *C*, which binds it firmly to the butt, also providing adjustment for wear. The key, which is of double wedge

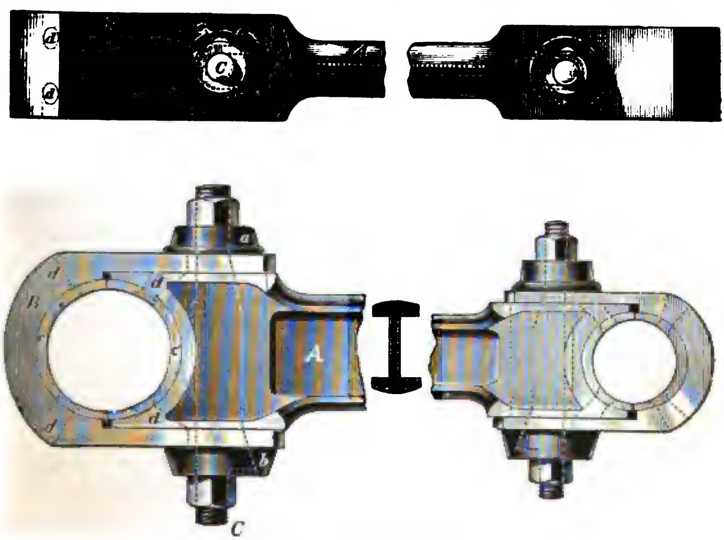

Fig. 28.—Cast-steel Rod.

form, bears against the strap at top and bottom in front, and against the butt end of rod on the back. The upward movement of the key will draw the wearing surfaces closer together. The washers, *a*, *b*, are cup-shaped and receive the tapering portions of the key. The butt and strap are bored out for the babbitt-metal bushings, *c*, *c'*, which are held in position by the babbitt anchors *d*, which are poured after the bushings are placed in position.

It will be noticed that the bushings or linings have no

flanges at the ends, the butt and strap being in width equal to the length of pin, and the bushings are therefore supported their entire length. This construction of rod is safer than the ordinary gib-and-key connection, especially when used at the crank-pin end. It possesses strength, rigidity, and lightness, important factors in successful operation.

In locomotive practice, solid-end parallel rods are found to be vastly less subject to breakage than the older forms fitted with straps. A broken end is comparatively rare when solid and bushed properly.

The Ratio of Length of Stroke to that of Connecting-rod is determined before the details of the design can be proceeded with; since this ratio determines the total length of the engine from cylinder to main journals and the proportions of every other piece of the machine. The longer the rod in proportion to stroke, the easier working the engine, but the greater the cost for a given power in consequence of the corresponding enlargement of every part; the shorter the rod, the less costly is the engine to build, but the greater the pressure on guides and pins, and the greater the friction and risk of heating and "cutting." Experience has led generally to the proportion of two or two and a half to one, the latter giving a long and easy-working rod, the former a rather short, but yet a manageable, one.

This variation of length of rod evidently compels a similar variation of length of every parallel member of the construction; and lengthening any rod means also enlargement of its cross-section and increase of weight and cost at a rapid rate. The tendency with builders is always to shorten up their engines to a minimum safe length, in order to decrease cost and to promote success in competition with the market. The maximum length of rod above given is usually considered a desirable proportion.

To compute the total stress, if P_1 be the force transmitted to the end of a connecting-rod, and θ be the angle between the direction of P_1 and the axis of the rod; then the force acting along the axis is $P = P_1 \sec \theta$. Hence, if a rod is n times

the length of the crank, the thrust along the rod, as a maximum, is $\frac{\sqrt{n^2+1}}{n} P_1$, or usually 1.03 to 1.08 P_1. In addition, the rod is subjected to strain due inertia. In slow-moving engines these may be neglected.

Suppose the crank of radius r. The crank-pin moves with a velocity V. Let w be the weight of the parts inclusive of half the weight of the rod. When the crank is near the centre, the resistance to acceleration is $\pm w \frac{V^2}{gr}$. Hence the thrust in the connecting-rod will be $P_1 - w \frac{V^2}{gr}$ at the beginning, and

$$P_1 + w \frac{V^2}{gr}$$

at the end of the stroke.

According to Grashoff, the bending action on the rod, due its inertia, is greatest at $\frac{6}{10}$ from the cross-head end, and that is the point at which the rod should be largest. The rod is tapered uniformly from the cross-head end to the crank-pin end, in quick-running engines, which is better than making it largest at the centre. Let w be the average weight of the rod, l, the length of the rod between the centres of the journals; r, the radius of the crank in inches; v, the velocity of the crank-pin. Then the greatest bending moment due to the swaying of the rod is shown by Unwin to be

$$M = 0.82 \frac{V^2}{r} \frac{wl^2}{g}.$$

The stress due to this moment is $f_t = \frac{M}{Z}$, where Z is the modulus of the section of the rod. Hence,

$$f_t = 4.92 \frac{V^2}{r} \frac{wl^2}{gbh^2} \text{ for a rectangular rod;}$$

$$= 8.351 \frac{V^2}{r} \frac{wl^2}{gd^3} \text{ for a round rod;}$$

d being the diameter, b the breadth at right angles to the plane of motion, and h the depth in the plane of motion.

Let G be the weight of a cubic inch of the material of the rod. Then, $w = Gbh$ or $\frac{\pi}{4} Gd^2$. Hence,

$$f_i = 1.28 \frac{V^2}{r} \frac{l^2}{gh} \text{ for a rectangular rod;}$$

$$= 1.712 \frac{V^2}{r} \frac{l^2}{gd} \text{ for a round rod.}$$

If P is the pressure acting along the rod determined as above, the stress due to that pressure is

$$f_t = \frac{P}{bh} \text{ for rectangular rods}$$

$$= \frac{4P}{\pi d^2} \text{ for round rods;}$$

And the total stress is $f_t + f_i$, which must not exceed the safe limit of stress.

The connecting-rod is thus seldom made less than twice, or more than three times, the stroke—four to six times the length of crank. A shorter rod produces irregular motion and intensifies the pressures on pins and guides, and on their support, too seriously, in consequence of its oblique action at mid-stroke; a longer rod becomes cumbersome, lacks rigidity, and compels the construction of too long and too costly an engine. The large figure is probably most commonly selected where the designer is not hampered in his plan, especially for high-speed engines.

The size of the rod at the middle should be computed as in compression, taking the rod as a long column, and, if its section is not circular, taking it as liable to spring in the direction of least stability. Since it is subjected to alternate, contrary, stresses, it must be calculated as if under approximately double stress, and the factor of safety thus doubled. Considered

as a simple column free at both ends, the principles of strength of materials make the load, as a maximum,*

$$P = \pi^2 \frac{EI}{fl^2};$$

when E and I are the modulus of elasticity of the metal and the moment of inertia of the section, and l the length in similar units. For a connecting-rod the load is to be taken as double, because of the reversed stresses, and

$$P = 2\pi^2 \frac{EI}{fl^2};$$

in which the factor of safety, f, should not be less than 8, ordinarily, or 6 as a minimum.

For a circular section, $I = 0.049\,d^4$, and for a rectangular section, $I = 0.083\,bd^3$. The value of E may be taken at 30,000,000, although this is a somewhat high value, and indifferently for both "mild" steel and good iron. Then, for a circular section,

$$P = \tfrac{1}{4}\pi D^2 p_1 = a\frac{d^4}{fL^2},$$

and, at the middle of the rod, we have

$$d = a' \sqrt[4]{fL^2D^2p_1} + C;$$

$$= a'' \sqrt{DL\sqrt{p_1}} + C;$$

in which a, a', a'', and C are constants determined by experience with the special class of engine to be designed. The Author has found successful practice to give values of $a''=0.01$ for engines of moderate speed, and up to $a'' = 0.015$ for fast engines and $C = \frac{1}{8}$ inch; D being inches, p_1 in pounds on the square inch, and L in feet from pin to pin. It may be advis-

* Materials of Engineering, vol. II, Iron and Steel; § 256.

able to correct the static load for the obliquity of action of the rod. This is done by multiplying that on the piston by the secant of the maximum angle of obliquity.

Thus for $D = 20$ inches, $L = 2$, and $p_1 = 125$,

$$d = 0.08\sqrt{DL\sqrt{p_1}} + \tfrac{3}{4} = 0.008\sqrt{20 \times 6 \times \sqrt{125}} + \tfrac{1}{2}$$
$$= 3.70, \text{ or } 3\tfrac{3}{4} \text{ inches, nearly.}$$

The same formula for an engine 60 inches diameter of cylinder, the rod 15 feet—"5 cranks"—in length, and the net pressure 60 pounds per square inch, gives

$$d = 0.08\sqrt{60 \times 15 \times \sqrt{60}} + \tfrac{3}{4} = 7''.48,$$

or nearly $7\tfrac{1}{2}$ inches. The piston-rods for the same engines are $3\tfrac{1}{8}$ and $7\tfrac{1}{8}$ inches in diameter.

The necks of the rod should be of ample area to sustain safely and with a factor of safety not less than 6 or 8, the *pull* to which it is to be subjected, and should also be computed as a column, the length of which is double the distance from the point at which the section is taken to the centre of the nearer pin; the larger of the two areas so determined should be taken as the proper section. It is rarely the custom, however, to make the neck of the connecting-rod much, if at all, smaller than the piston-rod.

15. The Crank and Pin should be designed with especial care, and it would probably be wise to give them both comparatively large factors of safety. Both these pieces are subject to various and widely varying stresses and are peculiarly liable to injury by shocks, due to looseness of bearings and to the introduction of water into the cylinder, which not infrequently result in their destruction, and occasionally that of the whole machine.

The Crank is sometimes made of cast-iron in stationary engines, but almost invariably of wrought-iron in other cases. The shape adopted depends greatly upon the general character

of the engine of which it forms a part. Small, fast-running engines are very generally fitted with a "crank-disk," a heavy disk nicely balanced on the shaft against the turning moment of the pin and so much of the weight of rod as tends to produce irregular motion of rotation.

The difficulty of forging large double cranks has led to the use of built-up cranks like that shown in Fig. 29 from Unwin. A double-collared, hollow, steel shaft is cut in half to form the

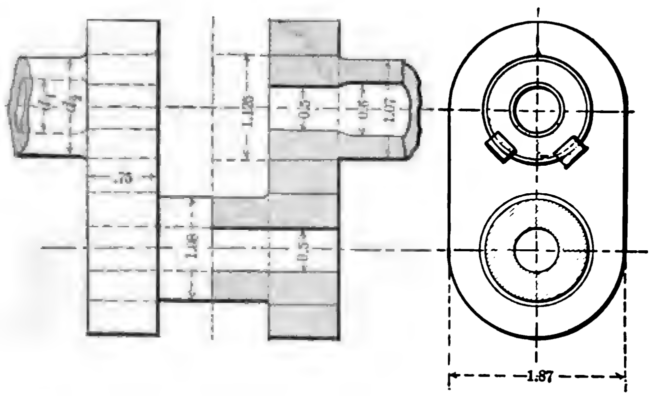

FIG. 29.—BUILT-UP CRANK.

single collared pieces. The crank cheeks or webs are forged solid, and then a small hole is bored at each end, and enlarged by being forged on a mandril placed on suitable supports; thus insuring that the metal is thoroughly worked. The cheeks are afterwards shrunk and keyed on the ends of the half lengths of shaft. The crank-pin is shrunk in, but is not keyed. The unit for the dimensions is taken as $\sqrt[3]{\dfrac{d_2^4 - d_1^4}{d_2}}$; d_1 and d_2 being the inside and outside diameters of the shaft.

Fig. 30 exhibits the proportions of a cast-iron crank designed by the Author for a slow-speed stationary engine of the Greene type, 18 inches in diameter of cylinder and 4 feet stroke of piston. The web is somewhat wider at the eye-end than usual.

The common form of cast-iron crank consists of a hub or

nave of about double the diameter of the shaft to which it is fitted, of an eye having similar relation of size to the crank-pin which it receives, and an intermediate connecting web. The depths of hub and eye are, respectively, equal to, or a little greater than, the diameter of the shaft and pin. The web has a breadth at either end of not far from three fourths the diameter of the adjacent hub or eye, and such thickness as to bear safely the stresses coming upon it, and reinforced by a

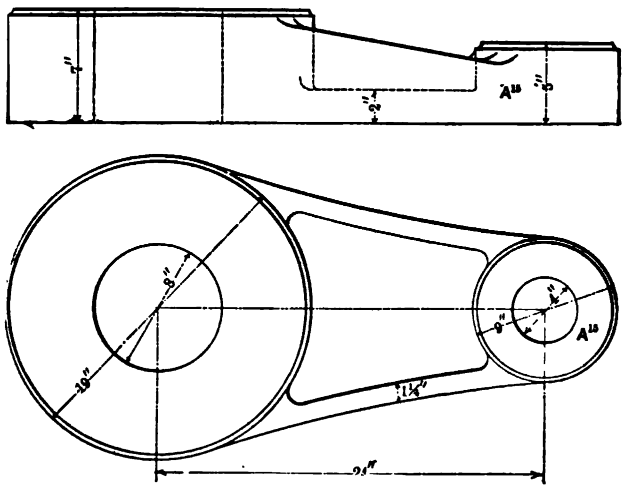

FIG. 30.—CAST CRANK.

flange or rim on each edge. The forged crank has hub and eye quite similarly proportioned; but it has its web made of rectangular section, or nearly so, and without flange, as the latter part cannot well be formed under the hammer. Where hydraulic or drop forging in dies is practised, however, the stronger shape may be obtained. In engines in which the shaft is fitted with bearings on each side the centre line of the engine, the crank-pin is carried between a pair of cranks, each in turn fitted to its section of the shaft. In many cases these cranks and the pin form parts of a continuous mass of iron or steel; in other cases, the shaft-section, cranks, and pin are all made separately and then united to form what is called a "built-up" construction.

The crank should be so proportioned as to be safe under its maximum load; and this maximum is attained when the steam reaches the piston at full pressure, the connecting-rod and crank being at right angles with each other. The pressure on the piston is

$$P = \tfrac{1}{4}\pi D^2 p_1,$$

and on the pin,

$$P' = P \sec\theta = \tfrac{1}{4}\pi \sec\theta D^2 p_1;$$

when θ is the angle of the rod with the centre-line of the engine. The twisting moment on the crank is

$$T = P'R = \tfrac{1}{4}\pi D^2 p_1 R \sec\theta;$$

R being the radius of the crank or the half-length of the stroke. The whole crank must be of sufficient strength to safely sustain this stress, and is computed as a beam fixed at one end and loaded at the other. In the case of the single or "overhung" crank, there is always a moment tending to twist the crank-web; but this is so small as usually to be amply covered by the common factors of safety, and is usually neglected. Then*

$$T = P'R = \tfrac{1}{4}\pi D^2 p_1 R \sec\theta = abd^2 \div l;$$

and as d is commonly fixed by the designer, as above,

$$b = \frac{\tfrac{1}{4}\pi f D^2 p_1 R \sec\theta l}{ad^2};$$

when f is the factor of safety, b, d, and l the thickness and width of the web, and the radius of the crank measured from shaft-centre to the point at which the dimension sought is to be measured. The empirical proportions adopted by builders will commonly be found to fall well within the calculated safe

* Materials, vol. II; Strength of Iron and Steel.

margin, as above. These proportions are, as obtained from the practice of successful designers, about as follows:

For the wrought-iron crank, the hub is 1.75 to 1.8 the least diameter of that part of the shaft carrying full load; the eye is 2.0 to 2.25 times the diameter of the inserted portion of the pin, and their depths are, for the hub 1.0 to 1.2 the diameter of shaft, and for the eye 1.25 to 1.5 the diameter of pin. The web is made 0.7 to 0.75 the width of adjacent hub or eye, and is given a depth of 0.5 to 0.6 that of adjacent hub or eye.

For the cast-iron crank, the hub and eye are a little larger, ranging, in diameter, respectively, from 1.8 to 2 and from 2 to 2.2 times the diameters of shaft and pin. The flanges are made, at either ends, of nearly the full depth of hub or eye. Cast-iron has, however, fallen very generally into disuse.

The crank-shaft is usually enlarged at the seat of the crank to about 1.1 its diameter at the journal. The size should be nicely adjusted to allow for the shrinkage or forcing on of the crank. A difference of diameter of one fifth of one per cent, 0.998 to 1.000, will usually suffice; and a common rule of practice gives an allowance of but one half of this, 0.001.

16. The Crank-pin is a journal subject to severe stresses and heavy friction, and is apt to give more trouble than any other journal in the whole system. It is best made of moderately hard steel, carefully turned nearly to size, and then ground exactly to gauge. Its diameter is determined by the magnitude of the maximum bending moment, and its length by the friction which it bears. Considered with reference to friction, simply, its diameter is a matter of indifference,* and its length is therefore the first thing to be determined. The diameter must be sufficient to insure against danger of springing or distortion, and the length must be such as to keep the maximum intensity of pressure at the highest speed of rubbing, and the product of the two quantities, well within safe limits. The pressure should, in the steam-engine, never, in any event, probably exceed 500 or 600 pounds on the square inch for

* Friction and Lost Work in Machinery and Millwork; § 29, 127, pp. 49, 239.

wrought-iron pins (35 to 42 kgs. per sq. cm.) or about twice that figure for steel. For marine and stationary engines the Author has, for many years, found a good length in inches of a steel pin to be

$$l = \frac{PR}{600{,}000};$$

in which P and R are the mean total load on the pin, in pounds, and the number of revolutions per minute. For locomotives, the denominator may be taken as 500,000. Where iron is used this figure should be reduced to 300,000 and 250,000 for the two cases taken. Pins so proportioned, if well made and well lubricated, may always be depended upon to run cool; if not well formed, perfectly cylindrical, well finished, and kept well oiled, no crank-pin can be relied upon. It is assumed above that good bronze or white-metal bearings are used.

Unwin gives the constants for the well-known formula,

$$l = a\frac{H.P.}{R};$$

as $a = 0.3$ to $a = 0.4$ for iron and for marine engines, and $a = 0.066$ to $a = 0.6$ for the case of the best steel and for locomotive work, where it is often necessary to shorten up outside pins as much as possible, and gives the following for values based on $a = 0.4$: *

VALUES OF l IN INCHES.

	R				
P	50	100	200	300	500
1000	0.2	0.4	0.8	1.2	2.0
1500	0.3	0.6	1.2	1.8	3.0
2000	0.4	0.8	1.6	2.4	4.0
3000	0.6	1.2	2.4	3.6	6.0
4000	0.8	1.6	3.2	4.8	8.0
5000	1.0	2.0	4.0	6.0	10.0
10,000	2.0	4.0	8.0	12.0	20.0

* Machine Design, § 84.

		R			
P	50	100	200	300	500
20,000	4.0	8.0	16.0	24.0	40.0
30,000	6.0	12.0	24.0	36.0	....
40,000	8.0	16.0	32.0	....	....
50,000	10.0	20.0	40.0	....	....

These give good working values.*

Since the length of the crank-pin will be directly as the power expended upon it, and inversely as the pressure, we may take it as

$$l = a\frac{IHP}{L};$$

in which a is a constant, and L the stroke of piston, in feet. Then the values of the constant as obtained by Mr. Skeel are about as follows:

$a = 0.04$, where water can be constantly used;
$= 0.045$, " " is not used generally;
$= 0.05$, " " " seldom used;
$= 0.06$, " " " never needed.

* The size of crank-pins required to prevent heating of the journals may be determined with a fair degree of precision by either of the formulæ given below.

$$l = \frac{P(V+20)}{44.800d}. \quad \text{(Rankine, 1865.)} \quad (1)$$

$$l = \frac{PV}{60.000d}. \quad \text{(Thurston, 1862.)} \quad (2)$$

$$l = \frac{PN}{350.000}. \quad \text{(Van Buren, 1866.)} \quad (3)$$

$$l = \frac{IHP}{130s}. \quad \text{(Skeel, 1873.)} \quad (4)$$

The first two formulæ give what are considered by their authors fair working proportions, and the last two give minimum length, for iron and for steel pins.

Formula (1) was obtained by observing locomotive practice in which great liability exists of annoyance by dust, and great risk occurs from inaccessibility while running, and (2) by observation of crank-pins of naval screw-engines. The first formula is therefore not well suited for marine practice.

Steel can usually be worked at nearly double the pressure admissible with iron running at similar speed.

Thus let, for each engine, the indicated power,

$$IHP = 1000;$$
$$L = 4 \text{ ft.};$$
$$l = \tfrac{1000}{4} \times 0.04 = 10 \text{ inches};$$
$$= \tfrac{1000}{4} \times 0.06 = 15 \quad \text{“}$$

for the minimum length and the best practice, respectively.

The Strength of the Crank-pin is determined substantially as is that of the crank. The load is usually assumed as carried at its extremity, and, equating its moment with that of the resistance of the pin,

$$\tfrac{1}{2}Pl = \tfrac{1}{32}t\pi d^3;$$

and*

$$d = \sqrt[3]{\frac{5.1Pl}{t}}.$$

Taking the maximum allowable stress on a square inch of section as $t = 9000$ lbs. per square inch for iron, the same authority obtains the following diameters:

VALUES OF d IN INCHES.

Ratio of $\frac{l}{d}$.

P	1.0	1.25	1.5	1.75	2	3
1000	0.84	0.94	1.03	1.11	1.19	1.45
1500	1.03	1.15	1.26	1.36	1.45	1.78
2000	1.19	1.33	1.45	1.57	1.68	2.05
3000	1.45	1.62	1.78	1.92	2.05	2.52
4000	1.68	1.88	2.05	2.22	2.37	2.90
5000	1.87	2.10	2.30	2.48	2.65	3.25
10,000	2.65	2.96	3.25	3.50	3.75	4.59
15,000	3.25	3.63	3.98	4.29	4.59	5.62
20,000	3.75	4.19	4.59	4.96	5.30	6.49
30,000	4.59	5.13	5.62	6.07	6.49	7.95
40,000	5.30	5.93	6.49	7.06	7.50	9.18
50,000	5.93	6.63	7.26	7.84	8.38	10.27

* Materials; vol. II. § 266.

For good steel pins these diameters may probably be safely reduced at least ten per cent.

17. **The Shaft and Journals,** which latter commonly determine by their proportions the whole shaft-design, are computed as of iron, generally; but the former is very commonly made of steel. Its office is the transmission of the varying torsional stresses of the crank to the balance- or fly-wheel, the resistance of the bending stresses of those efforts of the crank and of the weight of the wheel, and the support of those members of the structure accurately and rigidly in their intended locations. It must therefore be at once stiff and strong, rigid and safe. It should turn smoothly and steadily in its bearings, without heating and cutting of journals and with minimum friction. It must therefore have carefully proportioned, and well shaped and finished, journals, and well-fitted bearings, made of good material. The shaft should be as short between bearings as is possible, should be enlarged at the wheel and of minimum safe diameter at the journals; its bearings should be nicely turned and ground, to exact size and truth of form; and the crank and the wheel should both be carried just as close to the main journal as is practicable, in order to produce as little springing as possible under the transverse stresses thus caused by them. When thus designed, the transverse stress may be neglected as unimportant and well covered by the ordinary factors of safety.

Where the bending moment of crank or wheel is so great as to be considered, it is to be taken as related to the diameter of shaft by the formula*

$$d = \sqrt[3]{\frac{5.1 Pl}{f}};$$

in which P is the load on either journal, and l the distance to that journal; and for any point on the shaft, at the distance l' from the journal of which the diameter is d',

$$d = d'\sqrt[3]{\frac{2l}{l''}};$$

where l'' is the length of journal.

* Materials; vol. II. § 266.

The accompanying illustrations exhibit several common forms of steel shaft for single, compound, and triple-expansion engines.

FIG. 31.—STEEL SHAFT; DOUBLE CRANK.

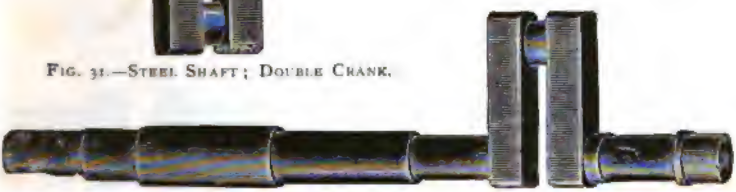

FIG. 32.—STEEL SHAFT; SINGLE CRANK.

A shaft subject to twisting efforts will have a strength proportional to the cube of its diameter; the torsional stress will be directly as the power of the engine "following" full

FIG. 33.—STEEL SHAFT; OVERHUNG CRANKS.

stroke, and inversely as the number of revolutions. We may therefore write, for minimum diameter of shaft,

$$d = a\sqrt[3]{\frac{H.P.}{R}}.$$

The value of a will somewhat exceed that obtained on the hypothesis of no transverse stress—in the proportion of from 1.10 in the most favorable cases to 1.5 in the case of the least.

The first is probably an ample allowance for all examples of good design. Unwin gives as values of a for the cases specified : *

For wrought-iron........................	3.293
For cast-iron..........................	4.47
For steel...............................	2.877

And they may commonly be taken as 3.5, 4.25, and 3, respectively. Mill shafting is commonly given higher values of a. Cranked axles of iron should be given $a = 4.5$, or more.

FIG. 34.—STEEL CRANK; "THREE-THROW."

Where the maximum thrust on the rod, making the twisting moment a maximum, is given as F, and that maximum moment, $M = Fl$, thus becomes known, we have†

$$Fl = M = 1.5708 sr^3 = 0.196 sd^3;$$

where s is the maximum shearing resistance, v and d are the radius and the diameter of the sections trained, and the diameter should be at least

$$d = \sqrt[3]{\frac{5.1 Fl.}{s}}$$

Molesworth uses, in shop practice,‡

$$d = \sqrt[3]{\frac{Pl}{K}};$$

in which the following values of K are employed:

Wrought-iron.........	1700	Cast-steel...........	2200
Cast-iron.............	1500	Bronze.............	460

* Machine Design; § 137.
† Materials of Engineering, vol. II, Iron and Steel; § 277, p. 526.
‡ Pocket Book.

P is taken in pounds; l and d are in inches. Cold-rolled shafting is sometimes used in small engines, in which case the value of K may be increased one half over that for common iron.

The following are the formulas approved by the British Board of Trade for the sizes of shafts for marine engines:

The main and tunnel and propeller shafts should be of at *least* the diameter found by the following formulæ:

Compound Engine with Two Cylinders.

$d =$ diameter of shaft in inches.
$D =$ diameter of high-pressure cylinder in inches.
$d' =$ diameter of low-pressure cylinder in inches.
$P =$ boiler-pressure.
$R =$ length of crank in inches.
$C =$ constant from following table.

$$d = \sqrt[3]{\frac{(D^2 \times P) + (d'^2 \times 15)}{C} R}.$$

Ordinary Condensing Engines with Two Cylinders.

$d =$ diameter of shaft in inches.
$P =$ boiler-pressure in pounds.
$D =$ diameter of cylinder in inches.
$R =$ length of crank in inches.
$C =$ constant from following table.

$$d = \sqrt[3]{\frac{D^2 \times P \times 2}{C} R}.$$

CONSTANTS FOR THE FORMULA.

Angle between Cranks.	Value of C for Crank and Propeller Shafts.	Value of C for Tunnel Shafts.	Angle between Cranks.	Value of C for Crank and Propeller Shafts.	Value of C for Tunnel Shafts.
90°	2,468	2,880	140°	1,858	2,168
100°	2,279	2,659	150°	1,806	2,108
110°	2,131	2,487	160°	1,772	2,068
120°	2,016	2.352	170°	1,752	2,045
130°	1,926	2,248	180°	1,746	2,037

Van Buren would make the diameters of iron engine-shafts

$$d = \sqrt[3]{bD^2pL};$$

and $b = 0.0025$ for single engines; $l = 0.0035$ for double, and $v = 0.005$ for triple engines; the diameter of cylinder, D, being in inches, the stroke, L, in feet, and the pressure, p, in pounds on the square inch. The values of a for steel are one half the above. The Author has successfully used the value, for iron, $b = 0.004$ for single engines and shafts, and $l = 0.005$ for a pair of shafts, as with the side-wheel marine engine.

A common and simple form of the expression already given is

$$d = a\sqrt[3]{\frac{I.H.P.}{R}};$$

in which a varies from 4.5 for heavy iron crank-shafts to 4 or less for lighter work.

The Journals of the main shaft should be proportioned with peculiar care. Next to the crank-pin journal, they are commonly the most troublesome journals in the whole engine. Their diameter should be calculated as above, as the minimum diameter of the shaft is commonly at the bearings. The length is then computed by the same rules as have been already given for journals in the preceding section; although they are often made, where practicable, still longer. With very powerful marine screw-engines, it will sometimes prove quite impracticable to secure a single journal at the crank end of sufficient length, if the crank be overhung, and a journal on each side the crank becomes an essential to good working. Thus, for a 21-inch shaft, on an iron-clad on board which the Author once served, the length of such a journal, as computed, should have been over four feet, a length quite impracticable to employ satisfactorily; since the shaft would spring in that distance sufficient to cause serious heating at the end of the journal next the crank. Such a case is very difficult of remedy. In

the instance referred to, a "spring bearing" supported on rubber cushions and two feet long was placed behind the 34-inch bearing carrying the shaft on the engine-frame.

The maximum allowable mean intensity of pressure may be, however, for all cases, computed, as in the case of the crank-pin or other journals, by the formula of either Professor Rankine or the Author. It must be borne in mind, however, that the friction-work on the main bearing next the crank is the sum of that due the action of the piston upon the pin, and that due that portion of the weight of wheel and shaft and of pull of belt which is carried there. The "outboard" bearing carries practically only the latter two parts of the total. In this, as in all other cases, the length of the journal will be proportional to the work to be done, and the crank-shaft journals will be made longer on the one side, and perhaps shorter on the other, than that of the crank-pin, in proportion to the work falling upon each, i.e., to their respective products of mean total pressure, speed, of rubbing-surfaces, and coefficients of friction.*

"*Brasses*, as the parts are called which form the surfaces of contact between journal and bearing, are made either of cast-iron or of brass, or bronze, or "kalchoid" † alloy. Where large, of ample area, and so strong or well backed as not to be liable to springing or fracture, they may be perfectly well made of cast-iron, the porous nature of which metal makes it readily absorptive of the lubricant, and causes it to work very satisfactorily. Oftener, especially in marine engines and in locomotives, these parts are made of bronze more or less approximating gun-bronze (copper 90, tin 10) in composition. When fitted to crank-pin and crank-shaft journals, they are often lined with babbitt or other "white-metal," to reduce friction and liability to heating. This expedient is less satisfactory of result on such journals as those of cross-head or

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* For values of the coefficient of friction under various conditions of operation, see "Friction and Lost Work in Machinery and Millwork," chap. VII.

† See Materials of Engineering; vol. III., Alloys.

rock-shaft, with which the motion is limited to a comparatively small arc of vibration ; its use on the crank-pin is considered by many builders as indispensable to the safe operation of high-speed engines.

The composition of the alloys used in brasses and their linings varies greatly. The bronzes so used generally range from copper 80, tin 20, to copper 90, tin 10, the more common being perhaps not far from 6 or 7 to 1, or about copper 85, tin 15. The strongest are the kalchoids, of which that of maximum possible strength, consisting of copper, tin, and zinc, was discovered by the Author,* and is composed of copper 54 to 58, zinc 44 to 40, and tin ½ to 2½, that having the composition copper 56, zinc 42 or 43, tin 2 or 1, being on the whole best. These alloys are extraordinarily strong, and, when not containing too much zinc and tin, amply ductile for bearings. British "naval brass" contains Cu. 62, Zn. 37, Sn. 1. Tobin's alloy is copper 58.22, zinc 39.48, tin 2.3, and belongs to this class. The British naval alloy (C. 62, Z. 37, T. 1) is another of the kalchoids, as is also the U. S. N. alloy for bearings (C. 88, Z. 2, T. 10).

The phosphor and manganese bronzes are extensively used. The first-named is any good bronze fluxed with phosphorus, usually by employing phosphor-tin as one of its constituents. The action of the metalloid is probably a purifying one, and the result of its introduction is to secure a sound and strong alloy. Lead is very generally used in brasses, and it is found to give a lower friction-coefficient and lessened liability to heating, when introduced in small proportion. Small quantities of iron, added, give increased strength, and especially if in the form of ferro-manganese in the proportion of 1 to 3 or 4 per cent. The Author's "maximum alloys," with a part of the tin replaced by iron, are given still greater toughness, and can be worked either hot or cold. Muntz-metal is one of the best of the pure brasses, and is composed of about 60 copper and 40 zinc.

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* Materials of Engineering; vol. III. p. 446, § 261.

"White-metal" for linings is composed, usually, of mixtures more or less closely approximating 70 tin, 25 antimony, and 5 copper, or of lead 75, tin 15, and antimony 10;* but they vary enormously, some containing two thirds copper, others two thirds antimony, with every imaginable proportion of zinc and tin. Zinc is often added to the bronzes for the purpose of securing soundness.

Standard practice is well illustrated by crank-shafts for marine engines, which have come to be made almost universally of mild steel, usually "open hearth" (Siemens-Martin) and often compressed. The British Board of Trade rule makes their diameter

$$=d\sqrt[3]{\frac{D^2 p_1 S}{a(2+R)}};$$

in which

D = diameter of low-pressure cylinder in inches;

p_1 = boiler-pressure, from vacuum;

S = stroke in feet;

R = ratio of volume of large to small cylinder;

a = constant = 170 for two cranks, at 90°; or 180 for three cranks set 120° apart.

The intermediate shafting has a diameter $0.95d$.

The crank-pin has usually a diameter and a length equal to d. Holes are often bored through shafts and pins for the double purpose of revealing flaws and to receive a reinforcing pin if needed. Where cranks are shrunk or built-up shafts, an allowance of 0.001 is made for shrinking. Gun-metal is the accepted material for bearings, and a lining of white-metal is commonly adopted, and is almost universally employed on large Atlantic steamers, both for crank-shafts and pins. Cast-iron main bearings, lined, are very serviceable. The thrust-bearing on the propeller or intermediate shaft is given an area of rubbing-surface equal to about 0.5 square inch per maximum indicated horse-power transmitted. The weight of the

* Materials of Engineering; § 141.

propeller and its shaft is taken on long bearings lined with strips of lignum vitæ, and kept cool by a small stream of water.

All important bearings subject to heavy work consequent upon exceptionally high speeds of rubbing, heavy pressures, or both; and especially if so situated as to be liable to be thrown out of line, should be lined with white-metal and, where practicable, as a matter of economy, have the brasses made of the best composition; though for rigid bearings of good area, cast-iron lined with babbitt or similar composition answers every purpose. For moderate speeds and pressures, ample area of cast-iron works well; that metal, especially if open in texture, absorbing oil and holding it with great effectiveness.

Large crank-shafts are now generally, in marine work, made hollow, to save weight and to insure the detection of internal defects. A 10-inch shaft bored out one half its diameter has about the strength of a $9\frac{3}{4}$ inch solid shaft and weighs 25 per cent less.

Where metals are run together under pressure there is supposed to be a film of oil always between the surfaces, as in the case of journals and slides. But since in practice this film often fails, care must be taken in the selection of materials for wearing-surfaces, so that "cutting" shall not occur when they come into contact.

Wrought-iron or steel, either soft or hard, runs well with wrought-iron or steel, soft or hard, or brass, or babbitt-metal, or malleableized cast-iron.

Wrought-iron or steel upon cast-iron is to be avoided if pressures and velocity of rubbing are high, and failure of lubricant possible. Cast-iron runs well with cast-iron.

As an illustration of the method of construction and fitting up of the crank-shaft and its accessories in a high-speed engine, where unusual care must be exercised, the accompanying design by Mr. Ide may be studied. The material is hammered steel.

The disk is bored, and forced on to the shaft with a hydraulic press, and then keyed. The hole for the crank-pin is bored after the disk has been put on, keyed, and finished.

The crank-pin is made of tool-steel, ground to gauge, and forced in with a hydraulic press under a pressure of 20 to 40 tons.

The crank-pin is hollow to the centre of the bearing, and the steel cap on the front is secured by a bolt, which is hollow

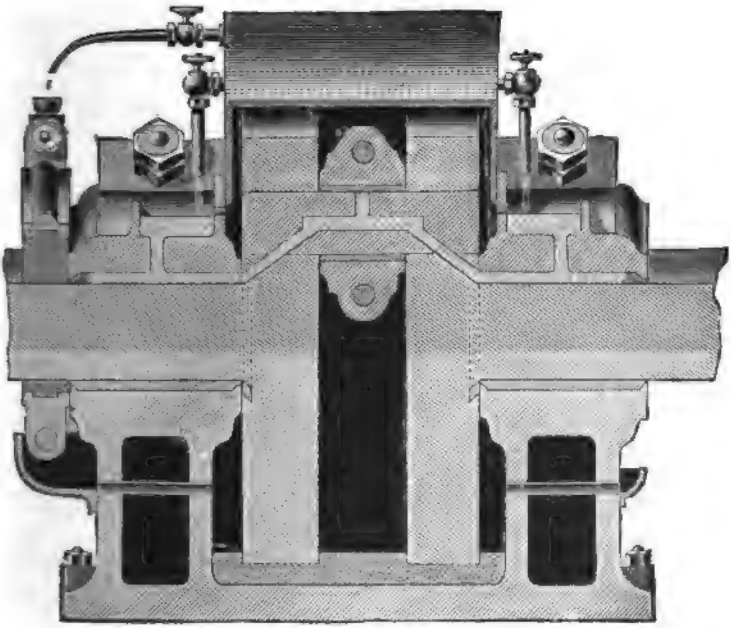

FIG. 35.—CRANK-SHAFT AND BEARINGS.

a portion of the distance. The projecting tube, with its hollow brass ball on the end, enters the cap, and connects with the oil-reservoir in the pin.

An oil-reservoir is cored out in the frame under the main bearing, to receive the waste oil. The oil from this chamber drains into a pocket cast under the crank-disk, on the front of the engine, from which it can be drawn off by a cock, suitably placed.

18. The Engine Bed or Frame, and its attachments, are designed to meet the requirements of each case, and are subject to no definite rule. Its office is to act as a brace, or

connecting member, between the steam-cylinder and the main bearing, in such manner as safely to carry the pull and thrust of the engine, and also, usually, to sustain the transverse stress thrown on the cross-head and guides by the angularity of the connecting-rod. It further, in many cases, carries the main-journal bearings, and the brackets or other attachments needed to support rock-shafts and other parts of the valve-gear. Whatever its form, it must be strong and rigid, and afford safe attachment to the other members of the structure, of which it is the backbone. A considerable degree of weight and solidity may be found as essential as strength in some cases.

In many designs, by Corliss and others, the frame is simply a brace between cylinder and main pillow-block; both of which latter are carried on the foundations and firmly bolted down; in other engines the frame is the engine-bed proper, and is secured to the foundation, sustaining the pillow-block and the cylinder; the latter being often, in horizontal engines, bolted to the bed by its front-head and overhanging, with no other support. The pillow block and the guides require firm support; but the only vertical effort affecting the cylinder being that of gravity, and comparatively insignificant, attachment to the bed in the manner indicated is quite sufficient.

Where the frame is a strut, it is sometimes of wrought-iron or steel; and the guides, in that case, commonly have independent support.

Computations of dimensions are rarely made. If the frame is heavy enough to safely receive its various attachments, and its thickness is made sufficient to satisfy the demands of the moulder and foundryman, it is usually perfectly safe against all anticipated stresses. Should doubt arise in any case, however, very careful investigation should be made to determine its strength.

The *Foundation* is commonly of brick, but sometimes, in the case of very large engines, of stone. It should be of sufficient depth to insure permanence of position, and so broad and solid as to be safe against injury by the jar of the engine. The "holding-down bolts" should be large, and should extend

to a good depth, the lower nuts being set in pocket-holes well toward the bottom and not too difficult of access. The main points of contact between engine-frame and foundation are usually made with heavy stones, through which the bolts pass.

Foundations should be so located and constructed that they shall not interfere with the operation, inspection, or repair either of the engines set on them or of adjacent machinery or plant.

They should commonly be built of good, strong, well-

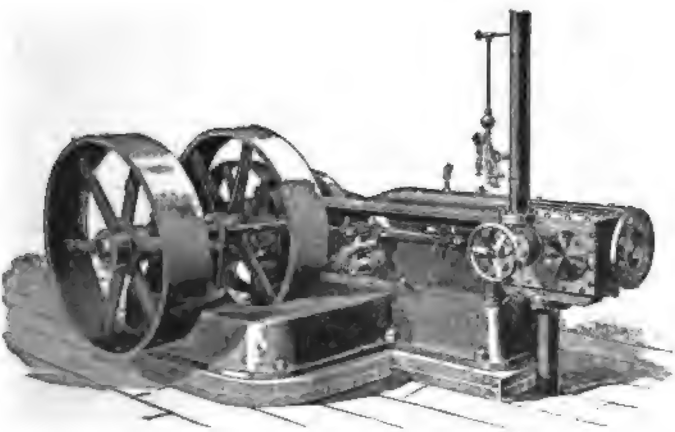

FIG. 36.—ENGINE WITH SUB-BASE.

burned but not over-burned brick, laid in the best cement,—never in mortar; and it is often wise to have a large and heavy sub-base of stone. Capstones are usually set on the top, and the engine mounted on these, the foundation-bolts running down through them to properly arranged pockets in the foundation, where the nuts and washers are set. Should the soil be soft or movable, piling should be resorted to in order to secure absolute certainty of permanence for the bed of the foundation. Where the engine is small and light, and well-balanced, it is permissible to place it on a floor, even in the upper part of a building; but it should be set on a heavy block of stone or a timber base well supported.

Foundation-bolts should be large, and carried well down into the foundation; the washers should be of good size, and the bolt-holes in the foundation are best made tapering, en-

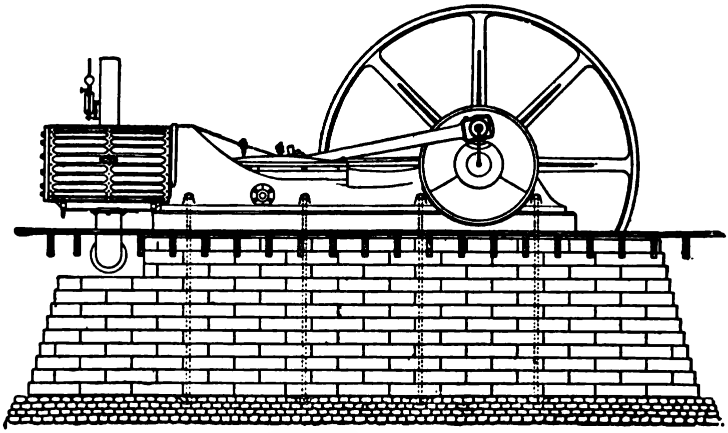

FIG. 37.—ENGINE FOUNDATION.

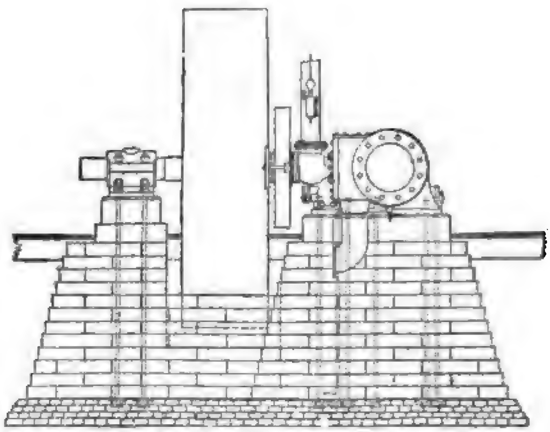

FIG. 38.—ENGINE FOUNDATION.

larged at the top to give space for the bolt to be set over should the holes in the engine-bed be found slightly out of line. When the engine is once set, cement, lead, or sulphur can be run in to fill the holes.

Many forms of engine-frames are illustrated in the various drawings of engines of different types and designs elsewhere given. In some cases, as seen in the figure here given of a Buckeye engine, a "sub-base" is used to give greater stiffness and structural firmness.

Here the engine is specially arranged, with two pulley-wheels, for driving a pair of "dynamos" by separated belts. The shaft is, by this system, held permanently in line, and the

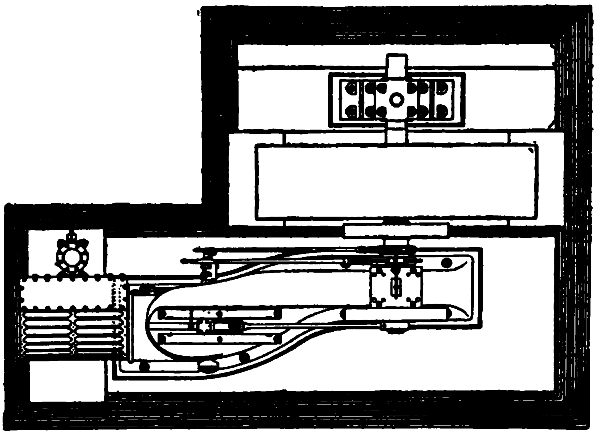

FIG. 39.—PLAN OF ENGINE FOUNDATION.

wear of journals and bearings is thus made comparatively uniform.

A good idea of a usual and satisfactory arrangement, and of the proportions of mill-engine foundations, is given by the three accompanying outline sketches of the plan, side-elevation, and end-view of a brick foundation for a stationary engine.

19. **Condensers** are employed on shipboard, invariably in ocean steamers, and very generally on river craft; they are less commonly adopted in stationary engines, and are not practically available for locomotive or portable engines, although the surface-condenser, with air as the cooling fluid, is sometimes used experimentally.

The action of the condenser is to transfer a portion of the heat contained in the steam exhausted from the engine to a

current of comparatively cold water and thus to convey it from the system and to effect the condensation of the steam. The quantity of heat so transferred is measured by the product of the weight of the steam condensed into the difference between its total heat in the state of vapor and that of the resulting liquid, measured from any convenient standard temperature, as the zero of either the Fahrenheit or the Centigrade scale. The quantity of water required to absorb this heat is measured by the quotient of its measure, in thermal units, by the difference in temperature between the original and the final temperature of that water. In the case of the jet-condenser, the final temperature of the water is that of the condensed steam, or the water of condensation, and is that of the vapor in the engine and of that in the condenser, and is also that of the air-pump. If the "total heat" of the steam, as given in the tables for the temperature and pressure of the exhaust, is h, the weight of steam w, that of the water w', and the initial and final temperature of the latter are t and t', the last being also the final temperature of the steam,

$$w(h - t') = w'(t' - t)$$

and

$$w' = \frac{w(h - t')}{t' - t}.$$

In surface-condensation, the steam is brought in contact with one side of the sheets, cooling-sheets or tubes, and the condensing-water with the other. The steam condenses and falls to the temperature of the metal, but the cooling-water does not; although it may approximate that temperature. In such cases, if the final temperature of metal and steam be t'',

$$w(h - t'') = w'(t' - t)$$

and

$$w' = w\frac{h - t''}{t' - t}.$$

The quantity of water demanded varies from about twenty-five or thirty times the weight of steam condensed, in the cooler seasons, to thirty or thirty-five with heated water.

This system is, for marine engines, or wherever the water-supply is very impure or acid, much to be preferred to the preceding, in which the water and steam mingle and the feed-water is contaminated by whatever injurious constituent may be present in the condensing-water. At sea it is found that

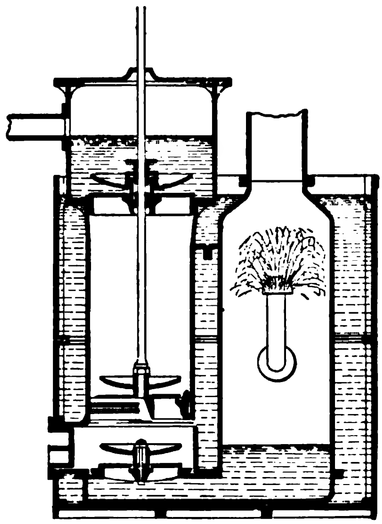

FIG. 40.—JET-CONDENSER AND AIR-PUMP.

the surface-condenser, while adding ten or fifteen per cent to the first cost of the engine, saves from fifteen to twenty-five per cent of the fuel as compared with engines fitted with jet-condensers, increases the durability of the boilers, if they are intelligently managed, very greatly, and gives some incidental advantages. The air-pump is made one half as large as with jet-condensers; but the necessary addition of a circulating-pump more than compensates that gain.

Common practice makes the volume of the jet-condenser from one third to one half that of the steam-cylinder; but the proportions should be made to depend upon the weight of steam

discharged into it at each stroke. It is made larger in small and fast engines. The surface-condenser is commonly composed of brass tubes of small diameter and considerable length (100 diameters often), set in an iron box of any convenient form. In marine engines the engine-frames often also do duty as condensers.

The total area of cooling-surface is commonly about one half that of the boiler-heating surface, with natural draught, and from $1\frac{1}{2}$ to 3 square feet per horse-power of engines. This cooling-surface is more effective if inclined or vertical than if horizontal, the difference often amounting to one third or more. A common rule is to provide for a condensing-water supply

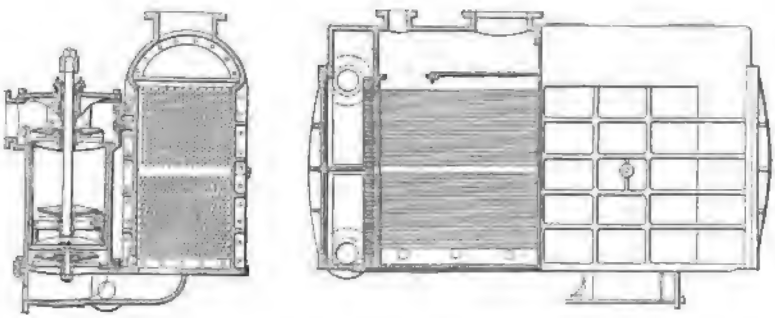

FIG. 41.—SURFACE-CONDENSER.

equal to at least 70 times that of feed-water. Common proportions of circulating-pump are from one twentieth to one thirtieth the volume of the steam-cylinder, the pump being single-acting. The volume for any given case may be computed by first ascertaining the volume or weight of water to be supplied, as above, and adding at least one third for waste and slip.

The general arrangement of a surface-condenser is shown in the accompanying figures. Here the exhaust steam passes around the tubes, and the water of condensation is removed by the air-pump shown.

It is very common to make the condensing-surface $\frac{2}{3}$ of the boiler-heating surface, and engines with surface-condensers are giving good results in which the condensing-surface in square

feet is from $2\frac{1}{2}$ to 3 times the indicated horse-power. From 50 to 120 pounds of steam are often condensed per hour, by each square foot of tube surface; but under conditions much more favorable than those which generally obtain in practice.*

The condenser-tubes are seamless drawn brass, tinned inside and outside. The tubes of Wheeler's are arranged in pairs, the smaller tube inside of the larger. The latter is thickened at one end, on which a substantial deep thread is chased. This end of the tube is screwed into a head of brass, and on the other end of the tube is screwed a cap, as shown. One end of the tube is also drawn thick, and a thread chased on it. This tube is also screwed into a head of brass. The tubes can be easily taken out and thoroughly cleaned.

Surface-condensers, as we have seen, are of two classes: (1) Those in which the steam is made to pass inside the tubes. (2) Those in which the cooling-water is circulated through the tubes. English naval practice adopts the first, while the merchant service usually adheres to the second.

In favor of the first class the following points are made: (1) The fatty matter is deposited on the front tube-plate and prevented from coating the tubes. (2) The large flat sides of the condenser are subject to the very slight pressure due to the head of water, and may be made light. (3) There is less hot surface exposed in the engine-room.

On the other hand, with the second class: (1) A large surface of metal is exposed to the steam. (2) The grease deposited on the tubes is easily removed. The softness of the deposit does not prevent the tubes being removed, as when scale from salt water is deposited. (3) The scale from sea-water can be

* Mr. John A. Tobin, U. S. Navy, in a report on the Improvements in Naval Engineering in Great Britain (Ex. Doc. 48, 47th Congress), says: "The proportion of condensing-surface to the horse-power in fifteen of the most recent (1883) types of high-speed merchant steamers by the best Scotch builders averages 1.95 to 1. This proportion, compared with the data of seven steamers, taken from a paper read before the Institution of Mechanical Engineers, by Sir F. J. Bramwell in 1872, having a ratio of 3.18 to 1, shows the saving effected in this direction during the past ten years."

removed without displacing the tube. (4) A better circulation of water is possible. (5) Economy of space and simplicity of design. (6) Tubes are better able to bear internal than external pressure. (7) Organic packing can be used; in the other form they are decomposed, and the products attack and destroy the tube ends.

Where several small stationary compound engines are used, one or more independent air-pumps, each with its jet-condenser, is an economical and compact arrangement. In this case the exhaust from the air-pump should be carried to the low-pressure receivers of the engines, so that the steam can be used expansively.

The jet "spray-pipe" and "rose-head" are still in use, although open to the objection of being easily fouled by dirt or materials in suspension in the injection-water. The cone system forms one of the best types, the advantage being that the thickness of the spray can be regulated by the adjustment of the cone. The condenser can be flushed by raising the cone from its seat. With high-speed engines, work the air-pump by belt connection, the pump being reduced to economical speed. In adopting the surface-condenser, with screw glands and packings of perishable materials, care must be exercised to keep the tubes tight, and to prevent grease from being pumped with the feed-water. A good filter should be used to take the grease out of the feed-water.

The advantages of the surface-condenser, according to Mr. J. F. Spencer, may be summarized as follows: *

(1) Freedom from injurious deposits in the boiler. This follows from using absolutely pure water, and not water that has been used for condensation. There is no necessity to scale the boilers or clean out salt.

(2) The boiler can be used with a higher pressure of steam. Scale and incrustations render it almost impossible to stay a marine boiler properly. Hence, when these evils are got rid of, we may use boilers of improved construction and higher pressure steam.

* Trans. Institution of Engineers (Scotland), 5th February, 1862.

(3) The foulest water may be used for condensation without risking injury to the boilers or engine.

(4) A more regular supply of feed-water can be relied upon. Under ordinary circumstances it requires constant watchfulness to regulate the feed and the brining.

(5) The load on the air-pump is more regular, so that in heavy weather the engineer need not reduce the injection-water.

(6) Fuel is saved, as no blowing-out is necessary. This saving of coal may often amount to from 15 to 25 per cent, which is something very considerable on a long voyage.

(7) Being able to use high-pressure steam, the economy of increased expansion can be fully realized.

(8) The boilers do not require cleaning so frequently; so that labor is saved, and there is less wear and tear.

(9) When no scale forms on the boiler, the iron plates more readily communicate the motion of the heat to the water; and thus fuel is saved from the absorption powers of the boiler being unimpaired.

(10) Reduction in size of boiler.

The disadvantages of a surface-condenser are:

(1) Additional pumps and machinery.

(2) Additional space occupied by the machinery.

(3) The use of the same water over and over again is held by some to corrode the boiler.

(4) Complication of tubes, etc.

(5) Liability to leakage.

(6) Increased first cost of from 10 to 20 per cent, and increased cost of repairs.

(7) More refrigerating-water is needed than for a jet condenser.

In large engines, the condensing system sometimes demands less than one per cent of the total power.*

The injector is somewhat modified to serve as a condenser. The figure shows this construction. The energy in the steam and water is found sufficient to carry all the water, air, and

* Trans. Am. Soc. M. E., June 1890.

uncondensed steam into the hot-well without the aid of an air-pump.

The cold water passes through A to the nozzle a, which is surrounded by the nozzles b and c, through which pass the exhaust steam by B and C from the two cylinders. Beyond the nozzles is a pipe P, which leads the current of fluid to the hot-well. The condensation of steam takes place between a and c. Thus the water enters A and rushes through a; it is met by steam at b, and the steam is condensed, but passes on, meeting more steam at c, and the steam, being all condensed, enters the hot-well by way of P.

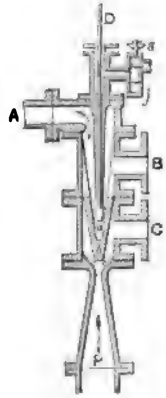

FIG. 42. — MORTON'S EJECTOR-CONDENSER.

The injection-water may be set in motion by a small jet of steam. The arrangement sj provides for this. The rod being screwed up, a jet of steam mingling with the water carries it forward to meet the exhaust. This jet is shut off after the apparatus is fairly at work.

Since all the calcium sulphate of sea-water is deposited when the pressure in the boiler exceeds about three atmospheres, the surface-condenser is a vital necessity in high-pressure marine engines. The weight of steam condensed per square foot in surface-condensers is usually about 15 pounds as a maximum. The rate of condensation varies with the velocity of flow of the condensing-water over the surfaces of the tubes. The allowance of area for the latest and most economical engines is not far from 2 square feet per I. H. P. Either type of condenser may be given any desired form. The only special precaution to be taken with the jet-condenser is to so arrange that the incoming streams of steam and water may be effectively mingled, and to make the volume of the condenser sufficiently large to prevent choking at starting, or flooding when in operation. The surface-condenser should be so designed that the tubes may not be clogged on the steam side by drops of water collecting from the condensing-steam. The more promptly and completely the surface is drained, the more

efficient the condenser. By introducing an artificial arrangement for continually sweeping it clean, Mr. N. Wheeler enormously increased its efficiency, and cases have been reported in which as much as 128 pounds of steam per hour has been condensed on one square foot of condenser-surface, or nearly one pound per degree of range of temperature between steam and injection-water; and Joule, by a system of concentric tubes, with the water between them, attained an equally high result. A surface maintained clean, and a high rate of flow of the condensing-water across it, give maximum results. The condensing-water should always enter on the surface nearest the point at which the water of condensation passes off, should leave it nearest the point at which the steam enters, and should have a rapid flow.

The packing of the ends of the condenser-tubes may be effected either by means of miniature, cheaply constructed stuffing-boxes, or by wooden packing of plain cylindrical form, three or four diameters long and an eighth-inch thick, filling the space between tube and tube-sheet, and holding the tube, and making a tight joint by swelling when the water strikes it. The latter is a very inexpensive device.

The use of a condenser will add about 10 pounds to the mean pressure in the cylinder; but this should not be accepted, however, as representing the gain. With high-pressure condensing engines, condensers will not be advantageous when the mean effective pressure is less than about 30 pounds. The condenser is useful, however, with low steam-pressure. With moderate pressures the condenser may effect an average saving of 25 per cent.

The amount of injection-water required is 1 to 1½ gallons per minute for each I. H. P.

The circulating-pump is often made separate from the main engine and driven by a small independent steam-engine. It is frequently made by adapting the common form of independent steam-pump to the work; but at sea it is probably oftener a centrifugal pump driven by one or a pair of small high-speed engines. This arrangement gives an independence of the mo-

tion of the main engines which is very valuable either when a heavy sea causes seriously great variations of its speed, or when, the engines at rest, it is convenient or important to hold the vacuum and thus to be prepared to start promptly. It gives also the advantage of ability to adjust the supply of water to the demand at any moment; and the uniformity of its resistances, and, consequently, of its motion, gives a higher mean efficiency than could otherwise be secured.

The independent surface-condenser, with its attached air- and

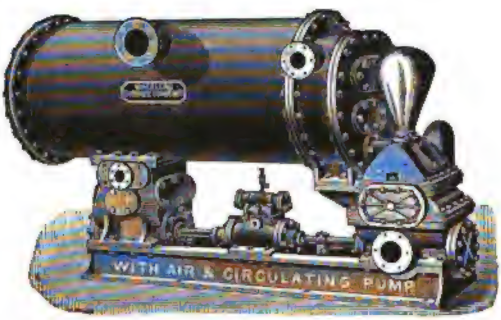

FIG. 43.—THE WHEELER CONDENSER.

circulating-pumps, has been found to possess so many advantages, especially for steamers in crowded harbors or on inland routes where liable to frequent and unexpected stops, that it is coming into more and more general use. It is quite unaffected by variations of speed, and but little by variations of load of the main engine; it permits the latter to be started and stopped suddenly, to be driven up to higher speeds, or to be reversed without embarrassment and without risk of bursting pumps, breaking heads, or danger of flooding the engine itself, with its attendant and serious risks.

The preceding figure gives a good idea of this type of condenser, as designed by Mr. C. H. Wheeler, and the succeeding illustration exhibits its internal construction and arrangement.

In this condenser, the exhaust-steam, entering the condenser by the nozzle A, comes in contact with the perforated scattering-plate O which protects the tubes from the direct im-

pingement of the steam. The steam passes out by the nozzle *B* to the air-pump.

The cooling-water is pumped into the compartment *F* through the nozzle *C*, and enters the small tubes as shown. After traversing the small tubes, the water returns through the annular spaces between the small and large tubes, and empties into the compartment *G*; from thence it passes into *H* by the passage *E*. The water then circulates through the tubes of the upper section, and finally passes out by the discharge-nozzle *D*.

When necessary to remove the tubes for cleaning or re-

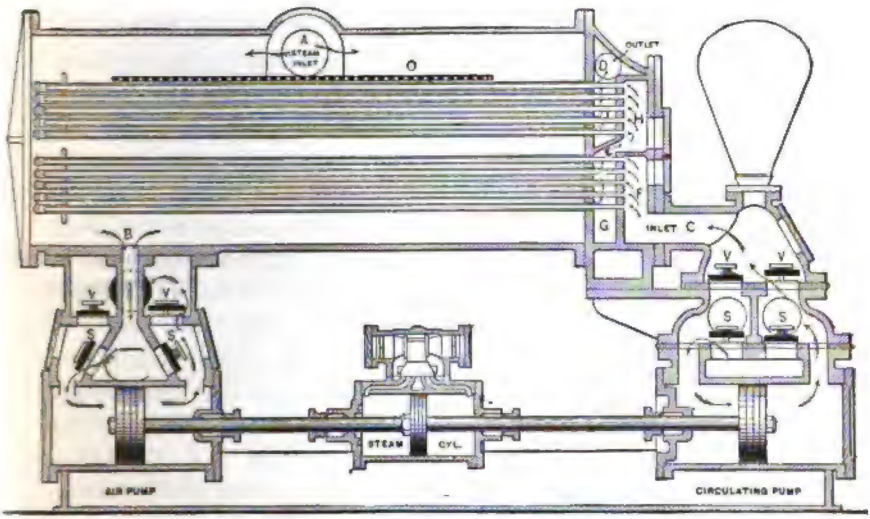

FIG. 44.—THE WHEELER CONDENSER.

pairs, both small and large tubes can be drawn out from the same end of the condenser. After removing the small tube the large tube is unscrewed and drawn through the hole left vacant by the screw head of the small tube—this hole being a little larger than the thick end of the large tube. It is necessary always to have the foot-valve of the air-pump lower than the bottom of condenser; otherwise it is impossible to drain the latter, in which case the condensed water will get chilled by contact with the tubes, and consequently supply cold feed

for the boilers. For economical engines, the weights of this class of machinery have been brought down to 8 and 10 pounds per horse-power.

In all such arrangements of independent pumps, the engineer endeavors to utilize the exhaust steam from their steam-ends, and all other steam otherwise lost, by employing it in suitable heaters to warm the feed-water. In this manner what might otherwise be an objectionable waste is reduced to so small an amount, often, as to become unimportant in the presence of the advantages derived.

The Theory of the Condenser of the steam-engine is exceedingly simple; but, in computing the quantities of heat to be carried away by the condensing-water, it must be remembered that it is the total heat of all steam drawn from the boilers, only reduced by the quantity converted into heat, and the wastes by conduction and radiation externally that must be calculated upon. Wastes by internal transfer to and from the metal of the steam-cylinder and the piston, or the water present, resulting in "cylinder-condensation," must be reckoned in the total quantity of heat taken from the boiler and passed into the condenser. This process of internal condensation, increasing, as it does, the amount of steam demanded by the engine by from 20 to 30 per cent, often, in that proportion compels a corresponding increase in the quantity of condensing water, in size and power of pumps, and in area of cooling-surface in the condenser. The methods of computation of its amount are given in Part I, and ample data for present purposes.

The following laws were deduced by Isherwood from his experiments where the thicknesses were $\frac{1}{8}$, $\frac{1}{4}$, and $\frac{3}{8}$ inch:

(1) The number of heat-units transmitted per hour through a square foot of surface is in direct ratio of the difference in temperature of the sides of the intervening metal.

(2) Within limits, the rate of transmission of heat through a metal wall is independent of its thickness.

(3) The thermal conductivity is as given in the following table:

Metal.	Thermal conductivity in terms of heat-units transmitted per hour through one square foot of material for a difference of temperture of 1° Fahr.	Relative thermal conductivity.*
Copper (refined)................	642.543	1.000 000
Brass, (60 Cu., 40 Zn.).........	556.832	0.866 607
Wrought-iron (best rolled)......	373.625	0.581 478
Cast-iron (several times remelted)	315.741	0.491 393

In Nichol's experiments it was found that: †

(1) The temperature of the water side of the tube is the arithmetical mean of the initial and final temperatures of the refrigerating agent, provided the rise in temperature is not greater than is found in ordinary surface-condensers.

(2) The efficiency of the condensing-surface is increased as the quantity of circulating water is increased.

(3) The surface is most efficient when the tube is horizontal.

(4) The number of heat-units transmitted through a unit surface in a unit time is greatest when the difference in temperature between the sides is greatest.

In designing, the formulas of Professor Whitham will probably be found to correspond satisfactorily with the results of practical experience; thus: ‡

The following assumptions are taken as warranted by experiment, viz.:

(1) The temperature of the steam side of the tube is uniform throughout its length (Joule), and the steam is saturated at a temperature corresponding to the reading of the vacuum gauge.§

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* Shock's Steam-boilers; p. 58.

† London Engineering; 20, 449.

‡ Trans. Am. Soc. M. E.; No. cclxxxv; vol. IX; 1888. Also Steam-engine Design; N. Y., Wiley & Sons, 1889.

§ This assumption is indorsed by C. Audenet in Étude sur les Condenseurs à Surface, and by E. Cousté in Annales du Génie Civil. See Van Nostrand's Engg. Mag., vol. I. Nos. 7, 9, and 10.

(2) The temperature of the water side of the tube is the arithmetic mean between the initial and final temperatures of the circulating water (Nichol).

(3) The conductivity of the surface is increased as the quantity of circulating-water used is increased (Nichol, Joule).

(4) The quantity of heat transmitted per hour through unit surface depends upon the difference between the temperature of the sides (Isherwood, Nichol); varies with the material used, and is independent of the thickness of metal (Isherwood, Joule).

Let $S =$ the condensing-surface in square feet;

$T_1 =$ the temperature of the steam in the condenser, or that of saturated steam corresponding to a pressure indicated by the vacuum gauge, in degrees Fahr.;

$T_2 =$ the temperature of the condensed steam as it leaves the condenser, i.e. the temperature of the hot-well;

$t =$ mean temperature of the circulating-water, or the arithmetical mean of the initial and final temperatures;

$L =$ the latent heat of saturated steam at a temperature T_1;

$k =$ perfect conductivity of one square foot of the metal used for the condensing surface for a range of 1° F., or 556.832 British thermal units for brass (§ 4, table);

$c =$ fraction denoting the efficiency of the condensing surface;

$q =$ rate of conductivity corresponding to a variable range of temperature $T - t$, and an elementary surface of rate ds; (T has a value between T_2 and T_1;)

$W =$ total number of pounds of steam sent to the condenser per hour, including all wastes of whatever kind.

The heat given up by the steam to the circulating-water is

$$\int q ds = W(L + T_1 - T_2). \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The range of temperature is

$$T_1 - t,$$

and

$$q = ck(T_1 - t).$$

The condensed steam at T_1 now gives up heat to the circulating-water, so that the range is at first

$$T_1 - t,$$

and finally

$$T_2 - t;$$

and at any instant while it is on an elementary area of rate ds, the range is

$$T - t.$$

Hence

$$q = ck(T - t).$$

Transforming equation (1) and integrating:

$$S = \frac{W}{ck}\left\{\frac{L}{T_1 - t} + \int_{T_2}^{T_1} \frac{dT}{T - t}\right\}, \quad . \quad . \quad . \quad . \quad (2)$$

or

$$S = \frac{W}{ck}\left\{\frac{L}{T_1 - t} + \log_e\left(\frac{T_1 - t}{T_2 - t_1}\right)\right\}. \quad . \quad . \quad . \quad (3)$$

The quantity $\dfrac{L}{T_1 - t}$ is always large, while $\log_e\left(\dfrac{T_1 - t}{T_2 - t}\right)$ is never greater than about 0.1. Hence equation (3) will be

practically correct, and much simplified, by dropping the last term, so that

$$S = \frac{WL}{ck(T_1 - t)}. \quad \ldots \ldots \quad (4)$$

The fractional coefficient, c, remains to be determined for a condensing-surface in ordinary use, i.e., coated with saline and greasy deposits. Data for determining the value of c is furnished by the experiments of Messrs. Loring and Emery,* as shown in the following table:

W	7261.54
S†	857.7

The value of ck being 180, equation (4) becomes

$$S = \frac{WL}{180(T_1 - t)}. \quad \ldots \ldots \quad (5)$$

This applies to an engine having an *independent* circulating-pump. When the pump is worked by the main engine, the value of S should be increased ten per cent. Formula (5) is recommended for use in designing.

In designing we may assume a vacuum not exceeding 25 inches of mercury. This corresponds to about 2.5 lbs. pressure. So that $T = 135$ and $L = 1020$; and equation (5) may be reduced to

$$S = \frac{1020W}{180(135 - t)} = \frac{17W}{3(135 - t)}. \quad \ldots \ldots \quad (6)$$

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* Report of Secretary of the Navy, 1874; Engineering, XXI. 121.

† The condensing-surface, though published as 900 sq. ft., has been found to be but 857.7.

The value of t will vary with the quantity of water used, and the season. It is about 60° F. in winter and 75° in summer. Since the larger value of t gives the greater value of S, take $t = 75$, and equation (6) becomes

$$S = \frac{17W}{3(135 - 75)} = \frac{17W}{180}. \quad . \quad . \quad . \quad . \quad (7)$$

The total number of pounds of steam condensed per hour is

$W =$ I. H. P. × pounds of steam used per I. H. P. per hour.

The circulating-water required per hour is, in pounds,

$$W_2 = \frac{W(L + T_1 - T_2)}{R}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where $R =$ the rise in temperature of the circulating-water.

The "Evaporator" or "Distiller" is designed to distil fresh water from sea-water without robbing the boilers, and yet to utilize the heat of the furnaces. The fresh water supplied by this machine may be used for drinking, or for replenishing that lost from the boilers by leakage.

In this apparatus, steam from the boiler is passed through a coil surrounded by sea-water, which latter is thus vaporized. The water condensed in the coils is returned to the boiler; the steam generated from the sea-water, if used for drinking, is treated with air by means of an air-injector, is condensed and filtered; but if used to replenish the boiler, the steam is conveyed directly from the evaporator to the main condenser, where it is condensed, and is supplied to the boiler *via* the air-pump, hot-well, and feed-pump. When used on a condenser, the evaporator vaporizes the water at a temperature below that at which any considerable quantity of scale is precipitated; the scale remaining dissolved in the dense water is extracted by a brine-pump.

There are two distinct forms of Baird's evaporator. Fig. 45 is a section through a vertical evaporator. Steam from the

boiler enters on the right-hand side, and the condensed water is discharged on the left into a steam-trap; the boiler-pressure may thus be kept in the coil, the trap permitting only water to be discharged. As both ends of the coil are secured to the lower head, the coil may be cleaned or scaled by breaking the joint on the lower head and lifting the shell off. As the transmission of heat through the coil surface is very rapid (about ten times as great as that in boilers), great precaution is essential against priming; the feed is therefore delivered in the central chamber, which causes a downward current in the centre; the steam is formed on the surface of the coil, and often lifts considerable water; much of this water flows into the central chamber directly, and the rest is delivered into the same chamber through the separator, as a dry pipe is always provided. The height of the water carried in the evaporator is sufficient to cover but half the coil, the designer having found that as much water is vaporized with this height as by covering the entire coil; the reason is that the upper half is kept wet by the foam, also because the water is a powerful absorbent, and much heat is conducted downward into the water. By lowering the water, the volume of steam-room is increased and some superheating surface is exposed, both of which tend

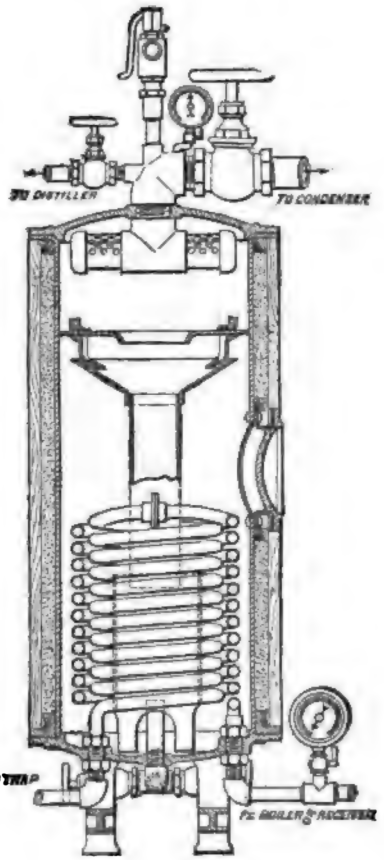

FIG. 45.—BAIRD'S EVAPORATOR.

to promote the supply of dry steam. The motion of a ship causes some little movement of the coil—which is but a spiral spring—and much of the scale breaks from its surface and falls to the bottom. Some height, however, is essential for taking the shell off the coil, and where this cannot be afforded the horizontal evaporator, Fig. 46, is used. This type has a num-

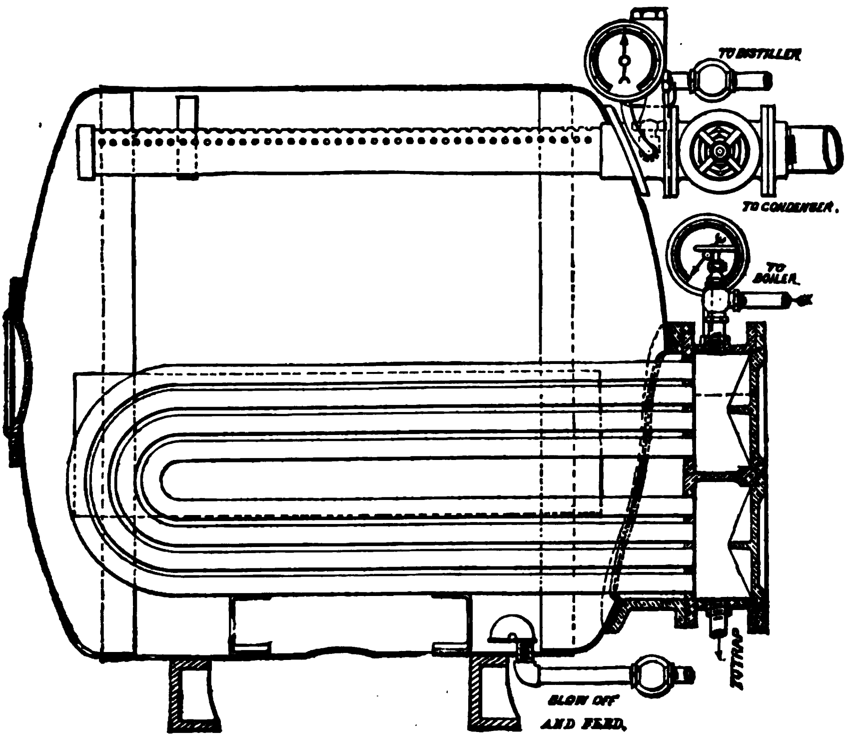

Fig. 46.—Evaporator.

ber of U-shaped tubes expanded into a cast-iron head. Steam from the boiler is forced into the upper chest and passed out of the lower chest into the steam-trap. This type has a larger surface of evaporation and more steam-room than the other, but is more bulky and is heavier. The shells are of thin plate steel, and the longitudinal seam is welded. The weight and bulk of an evaporator are much less than that of a "donkey-

boiler" of equal evaporative power, and as they are fireless generators, and operated at a lower temperature, they are rapidly coming into general use.

20. Pumps and Connections must always be designed with a liberal excess both of size and power. The essential elements, in this class, of a well-designed condensing engine are:

(1) The air-pump;
(2) The circulating-pump;
(3) The cold-water supply pump;
(4) The feed-pump;
(5) The "bilge-pump";
(6) Spare pumps of various kinds.

These pumps may be, any or all, either independent steam-pumps, or they may be driven by the main engine. They may be in regular use or intermittently employed,—essential parts of the "plant" or auxiliary.

The whole condensing system, including air and circulating pumps, is now often, and especially in marine work, made independent of the main engines on the score of convenience and safety; notwithstanding the considerable and inevitable enhancement of the cost of power expended in driving the pumps by the smaller engines thus used. The feed-pump, with all classes of engines, is usually attached to the main engine; but auxiliary feed-pumps, either plunger-pumps driven by an engine with fly-wheel, or what are technically known as steam-pumps, direct-acting pumps without fly-wheel, are supplied as auxiliary, to be used in case of accident to the main feed-pumps or when the engines are not working. The same remark applies to the "bilge-pumps" used on board ship to remove the water leaking into the vessel from the sea or from engines and boilers. Circulating-pumps are often centrifugal pumps; as they handle great volumes of water under small heads. Spare pumps are either intended to take the place of those regularly in use in case of accident or stopping them for repairs; or they are supplementary of the latter and used when

the main pumps are inefficient or are unequal to the demands upon them in times of emergency.

Air-pumps are either bucket or plunger pumps or bucket *and* plunger, as is seen in the various illustrations already given of the standard forms of engine. Its size must be sufficient, not only to dispose of the water to be removed from the condenser and all the air mingled with it, but also ample to meet the contingencies of a flooded condenser and large air-leaks into the engine or the condenser. The air-pump of the jet-condenser must remove all the condensing-water, plus all the water of condensation, plus the air entering with the feed-water and through leaky stuffing-boxes or joints. The air-pump for the surface-condenser is not required to handle the condensing-water, which, in that form, never enters the condenser; but it may be called upon to remove water leaking into the condenser from the water side.

No computation can precisely take into consideration these accidental, but often large, variations. Were there no such contingent liabilities, the latter case would demand a volume of pump but 15 or 20 per cent that of the pump for the jet-condenser; but the effect of these modifying conditions is, to an extent, to equalize the dimensions of the two pumps and, in practice, the latter is made from one third to one fifth the capacity of the former.

Since, in jet-condensation, the quantity of condensing-water usually exceeds that of the water of condensation nearly 30 times, we may adopt that factor. Then if

W_1 = weight of feed-water per I. H. P. per hour;
W_2 = that of the condensing-water;

then

$$W_2 + W_1 = 30\,W_1, \text{ nearly.}$$

If the density, or weight per unit of volume, of the water entering the pump be d, the volume will be, per hour,

$$V = \frac{W_1 + W_2}{d} = \frac{30 W_1}{d}.$$

If, in the air-pump,

$D =$ diameter,

$L =$ length of stroke,

$N =$ number of effective strokes per minute,

the volumes,

$$f 60 N \frac{\pi}{4} D^2 L = V;$$

in which f is a factor of safety, or allowance for contingencies, and the other quantities are "homogeneous."

The values of W_1 vary, in good practice, with engines of such size as are commonly made condensing, from 15 pounds in the most remarkably economical, to 30 pounds in ordinary work; d is usually, at the temperature of the condenser, not far from 60 pounds per cubic foot. Then

$$V = 30 \frac{W_1}{d} = 0.5\ W_1, \text{ nearly,}$$

$$V = 7.5 \text{ to } 15$$

in cubic feet per hour per horse-power; and, calling $V = 10$, $f' = 2$,

$$D = 0.3 \sqrt[2]{\frac{IHP}{NL}},$$

measured in feet, L also being in feet.

For the surface-condenser, in feet,

$$D = 0.15 \sqrt[2]{\frac{IHP}{NL}}.$$

Designers have been accustomed to make the capacity of the single-acting air-pump about $\frac{1}{8}$ that of the steam-cylinder of the simple engine, or of the low-pressure cylinder, if multiple-cylinder, for jet-condensers, about $\frac{1}{16}$ if surface-condensing, and one half these volumes if, as is now common, double-acting.

Valves for air-pumps are now usually of vulcanite, vulcanized caoutchouc, india-rubber containing about 3 per cent sulphur and a large percentage of zinc oxide; but they are liable

to serious injury if mineral oils are employed. Metal valves are more durable, though heavier and usually more costly if made of good bronze, as they should always be. The smaller their size, as a rule, the better. Their aggregate area should be large enough to restrict the velocity of flow to, at most, 400 feet per minute, thus often exceeding the area of the pump-bucket itself in the proportion of the velocity of the latter to the proposed minimum rate of flow.

Where no air-chamber is employed to equalize pressures, the volume of the pump should be somewhat increased. Ten per cent is a fair proportion for this increase.

The "modulus" or delivery-efficiency of such pumps ranges between 0.60 and 0.70, the latter being an exceptionally high figure, although, in some cases, a discharge equal above 0.95 the measured capacity of the pump has been attained by the use of exceptionally small and numerous valves and small clearance and port spaces. In such cases, the valves have been very light and held to their seats by slender coiled springs.

In computing the power demanded to drive the pumps, the friction of the pump itself, that of the water flowing at high velocity through its pipes and passages, and the resistances at the valves, must all be considered. The work performed by the pump will be as below, when

$Q =$ volume delivered per hour in cubic feet;
$W =$ weight of water so delivered in tons;
$D =$ diameter in inches;
$L =$ stroke in inches;
$N =$ effective stroke per minute;
$m =$ modulus of delivery;
$Q = 0.0026\, D^2 LmN$, nearly;
$W = 0.00006\, D^2 LmN$, nearly;

$$D = \sqrt{\frac{500\,Q}{LmN}}, \text{ nearly.}$$

Pipes should have such area as will restrict the velocity of flow of the water to a maximum of 500 feet per minute.

Circulating-pumps are usually either double-acting plunger-

pumps, with numerous valves, like the same type of air-pump and designed under similar rules; or they are centrifugal pumps. Its volume, in the former case, is usually not far from one half that of the air-pump. Its modulus is usually high, and with large pipes and valve-areas, at high speeds of engine, as with small screw engines, the Author has known the delivery to exceed the measured capacity of the pump, the inertia of the circulating stream making the velocity of flow nearly constant and at a rate between the mean speed of the pump and its maximum in its cycle. In such cases the delivery was always under water.

The valves are almost invariably of vulcanite. Both air- and circulating-pumps are lined with brass or gun bronze, and fitted with valve-seats of good brass. The pump-rod is cased with brass.

The centrifugal pump is made with gun-bronze wheel, or fan, and is driven at a speed of periphery exceeding, often, 500 feet per minute. It is rarely designed by the engineer planning the engines, but is commonly purchased of special makers of such pumps.

Steam-pumps of the direct-acting kind are also usually obtained in the same manner, the makers in such cases determining the dimensions.

A cold-water pump, to supply the condensing-water, as is sometimes necessary with stationary engines, must evidently have the capacity of the air-pump, as fitted to the jet-condenser. It may have any convenient form. *Bilge-pumps*, for marine engines, are usually simply a duplicate set of feed-pumps arranged to take water from out the bilge. Condensers are also fitted with a "bilge-injection," for a similar purpose, which has vastly greater capacity than that of the air-pump, or of the circulating-pump, and is used in emergencies. The size of this pipe is about two thirds that of the regular injection.

Feed-pumps are given the required capacity for supplying feed-water to the boiler, not simply at the mean rate of its evaporation, but with a factor of safety of 6 or 8; the latter figure being employed in marine engines with jet-condensers,

which compel the constant supply of an excess to compensate the loss by "blowing" to remove the accumulating salt from the boilers. A pair of pumps is usually employed, each with a factor of delivery of 3 or 4. Their capacity is commonly between 0.0025 and 0.005 that of the steam-cylinder, the former figure being adopted for the most economical classes of engine, the latter for good simple engines, the water needed being assumed to range from 15 to 30 pounds per I. H. P. per hour.

The pipes and valves should be proportioned to give a maximum velocity of flow not exceeding 400 feet per minute.

21. **The Water-supply and Delivery,** in the case of stationary engines, must usually be studied by the engineer with some care, with reference to both quantity and quality, the former, particularly, in the case of the adoption of the condensing-engine.

A water-supply must be of ample volume throughout the year; must be easily, cheaply, and certainly obtainable; must not be liable to interruption by reason of dry seasons, of variable demand by the engine to be supplied, or by the exigencies of other demands; must be free from either sediment or matter, either mineral or organic, in solution; must not be too greatly loaded with earthy matters in time of flood; and must not become too highly concentrated as a solution of lime or other salts in dry times.

Delivery to the boilers, and to the engine-condenser, must be insured with absolute certainty; and alternative sources and methods of supply are always desirable. The pumps or other mechanism of supply must be made as safe as is possible, by every precaution against their failure and, in all important work, spare pumps and connections should be at all times ready for use. This should be scrupulously considered and provided for in the original design of the "plant." All that is here stated is especially true in regard to the feed-water supply for the boilers, the failure of which, at any time, may prove disastrous.

Pumps and pipes should be of more than ample capacity, and should usually be capable of giving, in times of emergency,

from 4 to 8 times the ordinary supply. The workmanship of the whole system should be looked to with great care, and the dangers arising from depreciation, usually very rapid, should be constantly kept in mind and provided against in every possible way.

22. **Minor and Miscellaneous Details** of design need not be here considered at length. They are illustrated in every engine in operation and in every set of drawings of engines of whatever type. In many cases their forms and proportions are determined by precisely the same rules as larger parts.

Thus: the links and rods of the valve-motions and the starting-gear have the same general forms as the heavier connecting-rod; the rock-shafts are subject to the same rules of design and construction, substantially, as the crank-shaft; the pins about the engine have to do either such work as the crank-pin, or to act as bolts or rivets, subject, usually, to transverse shearing stresses.

All such parts may be readily designed when the kind and magnitudes of the stresses to which they are subject are ascertained or given. It is sometimes, however, difficult to analyze them in which case the doubt is met by the adoption of high factors of safety.*

The importance of securing dry steam at the engine, both economically and as a matter of safety, is exceedingly great. Wet steam in any engine working with a high ratio of expansion exaggerates internal wastes greatly; in jacketed engines, it may render the jacket comparatively inefficient, even if it does not entirely destroy its value; and fast-running engines are especially liable to wreck by the entrance of primed water into their cylinders. Marine engines are peculiarly liable to this kind of accident in consequence of the surging about of the water in their boilers with the pitching and rolling of the ship in a seaway.

It is thus found advisable, in most cases, to provide for the separation from the steam, as it enters the engine, of any "en-

* For such details, see special treatises on the subject, such as Unwin's Machine Design, Whitham's Steam-engine Design, and the better class of engineers' "pocket-books" and handbooks.

trained" water. This is effected by the use of "separators" placed at the junction of steam-pipe and steam-chest, or as nearly so as practicable. These act by either of two processes: expansion of the current, securing decreased velocity with increased cross-section, and thus permitting the suspended water to settle out of the steam; and centrifugal whirling of the current, forcibly throwing the particles of water out of the current against the sides of the separator, to settle into the lower portion of its chamber; whence it is trapped or carried off by a "steam-loop" and returned to the boiler. A separator is now very generally used on all classes of engine; it is also sometimes introduced between the cylinders of the compound or other multiple-cylinder engine, or in advance of a superheater to reduce the work of the latter. With hard-driven boilers it is especially desirable. Incidentally, it is useful as a dirt- and grease-extractor.

The illustration exhibits a form of separator devised by Mr. J. P. Stratton, in which the two processes just described are combined. The large chamber below permits settling to occur effectively; while the entrance of the wet steam is arranged at the top in such manner as to compel it to take the path indicated by the arrow, and to thus make effective the centrifugal action already alluded to. A water-gauge glass permits the attendant to see how much water is present, and the drain-pipe at the bottom takes it away.

FIG. 47.—THE "SEPARATOR."

Even with a separator in the steam-pipe, there exists a liability, at times, with small or badly managed boilers, to the passage of such large masses of water into the steam-pipe that they cannot be completely disposed of and may enter the engine. To meet this contingency, the use of relief-valves is usually resorted to in marine engines. The "breaking-cap" is introduced in some

forms of high-speed engines. That here shown is the design of Mr. Ide.

In the event of flooding of the engines from the foaming of the boilers, they protect it by bursting and thus allowing the water to escape.

In the case of the bursting of a safety-cap, it can be replaced quickly and at trifling expense.

Systems of lubrication, as may be seen on examining the illustrations of the many forms of engine distributed through

FIG. 48.—BREAKING-CAP.

the book, differ as greatly as do the main designs themselves. In all cases they should be convenient, safe, and sure. But little difficulty is experienced, usually, except with crank-pin and governor-journals. As is elsewhere seen, these are now often worked in a bath of oil or of oil and water, and free and safe lubrication thus insured. A good and usual device is seen in the next figure, which represents the centrifugal crank-pin-oiling device first introduced in the Buckeye Engine, now very widely adopted. It consists of tube *b*, the inner end of which is terminated by a brass internally chambered cup or ring *a*, which is concentric with the shaft, and its outer end is rigidly secured to the crank-pin in such manner that its internal passage is in

communication with a passage in the pin emerging at *c*, as shown by the dotted lines, so that oil introduced at *a* promptly issues at *c* and effectually lubricates the boxes. An ordinary feeder-cup *d* with needle-valve at *e* to regulate supply, the whole mounted on stand *f*, furnishes the supply of oil required while the engine is running.

FIG. 49 —CRANK'OILING.

The tube leading from valve *e* to the ring-cup *a* is bent at a right angle downwards inside the ring-cup. The chamber in *a* is made large enough to embrace the bent end of the tube without at any time touching it, and to allow the drops of oil to be clearly seen as they fall; and also to admit oil to be introduced by hand at any time without disturbing the adjustment of the feed from the cup.

In cylinder-lubrication, the various forms of "sight-feed" apparatus are most used, as supplied by makers; and an oil-supply of good quality, to the amount of about a drop a minute

for each 1000 square feet of surface to be lubricated, will usually suffice with moderate speed of engine. The quantity should be increased with increasing speeds, and pressures exceeding 100 pounds per square inch.

23. **Belting and Gearing** are both in common use for transmission of power from the engine to its work; but the much lower cost of the former, its greater convenience, its ready adaptability to any distance between main and jack shaft, or line-shafting, and driven machine, have given it the foremost place, and gearing is continually less and less used. The latter is also very liable to cause serious damage if even a chip is broken from a tooth, or if any other substance falls into its engaging cogs. The breakage of a belt is rarely a serious matter. Where slipping cannot be permitted, gearing must be used.

The material of belts is commonly leather, often rubber or canvas, or both, and occasionally other substances. Endless belts, where they can be employed, are preferable to laced or otherwise connected belts. The strength of the solid leather in belts is 2500 to 5000 pounds per square inch; at the lacings, even if well put together, only about 1000 to 1500. If riveted, the joint should have half the strength of the solid belt.

Working stresses are commonly from 150 to 300 pounds per square inch of section, or 50 to 100 pounds per inch width of single belting. An old rule allows a horse-power to one inch width of single belting running 1000 feet per minute. Generally, we have

$$H.P. = \frac{PV}{33,000};$$

where the net pull, P, and the speed, V, are in pounds and feet per minute. This gives a very liberal allowance. A "double belt" should give a half more power, at least, than the single belt.

Large engines driving mills are often fitted with a pulley fly-wheel of several feet width of face, and arranged to carry two, or even three, wide double belts. In some such cases,

the speed is driven up to a mile a minute. The pulley-face is usually "crowned" or made higher at the middle to insure the belt against running off. A flange is sometimes used at the edge of the pulley for this purpose; but it is only effective with narrow and thick belts, and the other device is commonly much safer. Where two or more belts are carried on a pulley-face, they should be set well apart and each on its own crowned section of the rim. Where a main belt connects engine with a head-shaft, or jack-shaft, and another belt drives a machine or another shaft from the latter, the one belt is sometimes run directly on top of the other.*

The following table will serve all ordinary purposes:

H.P. OF BELTS—UNWIN.

Velocity of Belt, ft. per sec.	Width of belt in ins. $\frac{7}{32}$ inch thick when the horses' power transmitted is									
	1	2	3	4	5	7½	10	15	20	25
1	15.7	31.4	47.0	63.0	....	....	....	....	....	....
2½	6.3	10.6	18.8	25.2	31.2	46.8	....	....	....	....
5	3.1	6.3	9.4	12.6	15.6	23.6	31.4	47.2	....	....
7½	2.1	4.2	6.3	8.4	10.4	15.6	21.0	31.2	42.0	52.4
10	1.5	3.2	4.7	6.4	7.8	11.8	15.7	23.6	31.4	39.2
12½	1.3	2.5	3.7	5.0	6.4	9.4	12.6	18.8	25.2	31.2
15	1.1	2.1	3.1	4.2	5.2	7.8	10.5	15.6	21.0	26.2
20	.79	1.6	2.4	3.2	3.9	5.9	7.9	11.7	15.7	19.6
25	.63	1.3	1.9	2.6	3.1	4.7	6.3	9.4	12.6	15.6
30	...	1.1	1.6	2.2	2.6	3.9	5.2	7.8	10.5	13.1
35	...	...	1.3	1.7	2.2	3.4	4.5	6.8	9.0	11.2
40	...	...	...	1.5	2.0	2.9	3.9	5.9	7.8	9.8
45	...	...	...	...	1.8	2.6	3.5	5.2	7.0	8.8
50	...	...	...	...	1.6	2.4	3.2	4.7	6.3	7.8
60	...	...	...	...	1.3	2.0	2.6	3.9	5.2	6.5
70	...	...	...	...	1.1	1.7	2.2	3.4	4.5	5.6
80	...	...	...	...	...	1.5	2.0	2.9	3.9	4.9
90	...	...	...	...	...	1.3	1.8	2.6	3.5	4.4
100	...	...	...	...	...	1.2	1.6	2.4	3.1	3.9

According to Achard, when the belt is wide a partial vacuum is formed between the belt and pulley at a high velocity.† The

* For the theory of the friction of belting, see "Friction and Lost Work in Machinery and Millwork, § 31, 118.

† Proc. Inst. Mech. Engrs., 1881.

pressure is then greater than that computed from the tensions in the belt, and the resistance to slipping is greater. This has the advantage of permitting a greater power to be transmitted by a given belt, and of diminishing the strain on the shafting. For heavy and fast belts, the speed is from 4000 to 6000 feet per minute. The widths are such that the driving-pressure P is 50 to 67 pounds per inch width of the belt, and the greatest strain in the belting is about 156 to 185 pounds per square inch of section. The belts are generally not more than $3\frac{1}{2}$ to 4 feet wide.

24. **The Valve-gear and Regulation** are, all things considered, those elements of the design of the steam-engine which demand of the designer the most careful consideration and the most deliberate judgment. The economical performance of the engine and the smoothness and uniformity of its motion, a matter of, often, no less importance, are determined by the character of the system of steam-supply and distribution. A simple, three-ported, plain slide-valve is at once the least costly and the least efficient valve. The system of placing a steam- and an exhaust-valve at each end of the cylinder, four valves altogether, is the most expensive and the most economical. Regulation by throttling the steam at the steam-pipe or steam-chest, and that which effects its purpose by fixing automatically, and at each stroke, the point of cut-off and ratio of expansion, are, similarly, the two alternative systems. Of these the latter is preferred.

A "detachable" or "drop" valve-gear can be used on slow or moderately fast engines; but a "high-speed engine" must have a "positive-motion" valve system. Where fuel is cheap, the less costly but more inefficient systems are employed; but where fuel is costly, the engineer finds it wise to go to any necessary expense to secure an economical system. It thus happens that engines of the Corliss, Greene, and allied types are generally in use for mill and general purposes; while the fast-running, simpler forms of engine have come into very extensive use for driving electric-lighting machinery. The locomotive is thus made as simple in design as possible; while the

steam pumping-engine is often built for economy, with little regard to cost. This subject is of such importance that a separate chapter will be devoted to its consideration.

25. **Regulating Apparatus,** including governors and fly-wheels, and involving the effects of the inertia of irregularly

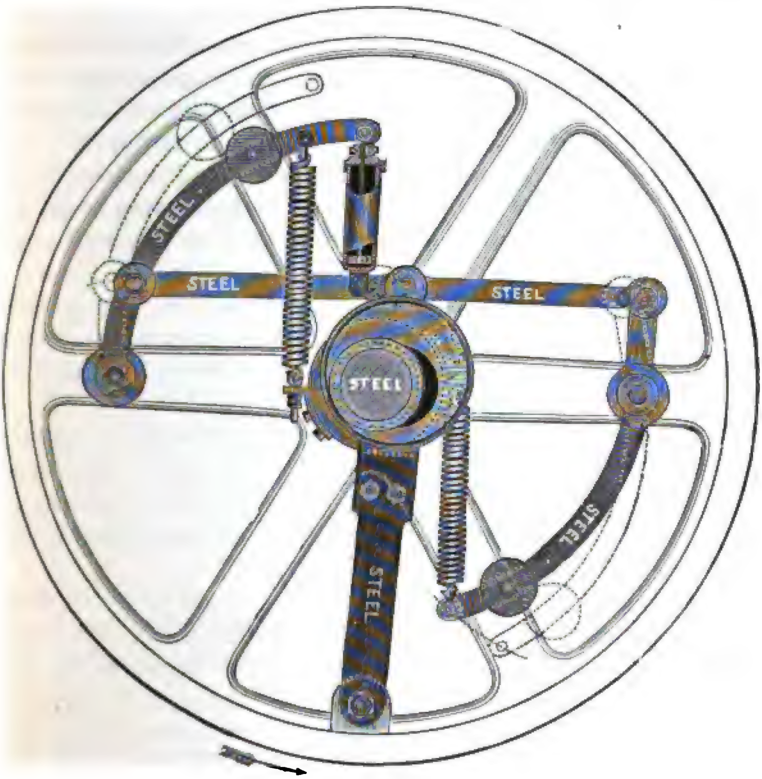

FIG. 50.—DETAILS OF GOVERNOR.

moving parts, must be considered in a chapter (III) specially devoted to them, as the subject is extensive as well as important, and demands a somewhat extended and detailed study.

As will be seen later, the regulation of the engine includes the reduction of all irregularities in the motion of its parts to perfect uniformity or to smooth harmonic motion, whether considered with reference to long or to short periods of time, or to

successive cycles or parts of a single revolution. The problem resolves itself into two principal parts: (1) to prevent irregularities due to variation of work demanded of the engine, by proper design, construction, and adjustment of governor and fly-wheel; (2) to reduce to a minimum all variations of the turning moments taking effect at the crank-shaft, due to irregularities of effort at the piston, to variations introduced along the line of transmission by the inertia of reciprocating parts, and those due to the action of the unsymmetrical kinematic chain necessarily employed. A sensitive, powerful, and synchronous governor, ample fly-wheel energy, and proper adjustment of weights and counterbalancing in the moving parts, are the essentials to securing uniformly steady motion and exact speed.

The method of distribution of material and the general design of the shaft-governor is well shown in the accompanying design, Fig. 50, by Mr. Ide, the metal used being indicated on each part.

The free use of steel and the simplicity of construction are notable points in good standard work. The sketch also shows the method of attachment of the governor to insure smoothness in operation and freedom from oscillation of weight and corresponding variation of engine-speed.

26. Peculiar Types of Engines are found to be, in some cases, especially well fitted to special work. Thus, the introduction of the various applications of electrical energy has led to the development of the whole class of "high-speed" engines. Among these is the "steam-turbine," either a common direct-flow turbine or a "scroll-wheel" of the Whitelaw type, in which the reaction of the current develops an amount of energy which approaches the ideal maximum as the speed of rotation and of the discharge orifices increases, within practical limits. The design of the former, as of nearly all peculiar forms of engine, is a subject of no great difficulty, all the principles which have been given applying to them as to ordinary engines. The design of the latter type involves merely the determination, first, of the maximum speed of rotation practi-

cable, and next, the diameter permissible. These points settled, the wheel is given a general form such as will least affect efficiency by its friction resistances, either on its journals or in the air, and a size of orifice such as will discharge the required weight and volume of steam at the pressure fixed upon. In some cases, compounding has been resorted to, to reduce the speed of rotation for high efficiency, with good results. Properly designed and operated, these machines should give quite as high efficiency as the small engines which they are intended to compete with.

The ideal efficiency of these wheels is, as shown by Rankine,*

$$E = \frac{2\,ar}{\sqrt{2\,gh}\left(\frac{ar}{2gh} + \sqrt{1 + \frac{ar}{2gh}}\right)};$$

where a is the angular velocity, r the radius to the orbit of the orifice, and h is the height due the velocity of outflow of the fluid, and E approaches unity as the value of ar approximates infinity.

It is an interesting and curious fact that this earliest of all steam-engines, antedating James Watt nearly two thousand years, should have as high an ideal efficiency as the best of modern engines.

A common speed is that which makes the velocity of orifice equal $v = \sqrt{2\,gh}$, that of exit of the steam.

The speeds of the steam-turbines enormously exceed those of any form of engine with reciprocating piston, or even of the so-called rotary engines. The three- and four-cylinder engines of the Brotherhood type, in which the several cylinders are usually grouped radially about a common crank and shaft, often exceed 1000 revolutions per minute and have been driven, experimentally, above 2000; but the steam-turbine of Parsons makes 10,000 and even 20,000 revolutions, and the Dow turbine is reputed to have attained 25,000.†

* Steam-engine; p. 196, § 176.

† Trans. Am. Soc. Mech. Engrs.; vol. x, 1889, p. 681.

In many cases it is found wise to make a peculiar or a special design to meet a specific case. Thus, in the accompanying illustration, we have a directly-connected pair of high-speed engines driving a low-speed dynamo-electric machine on

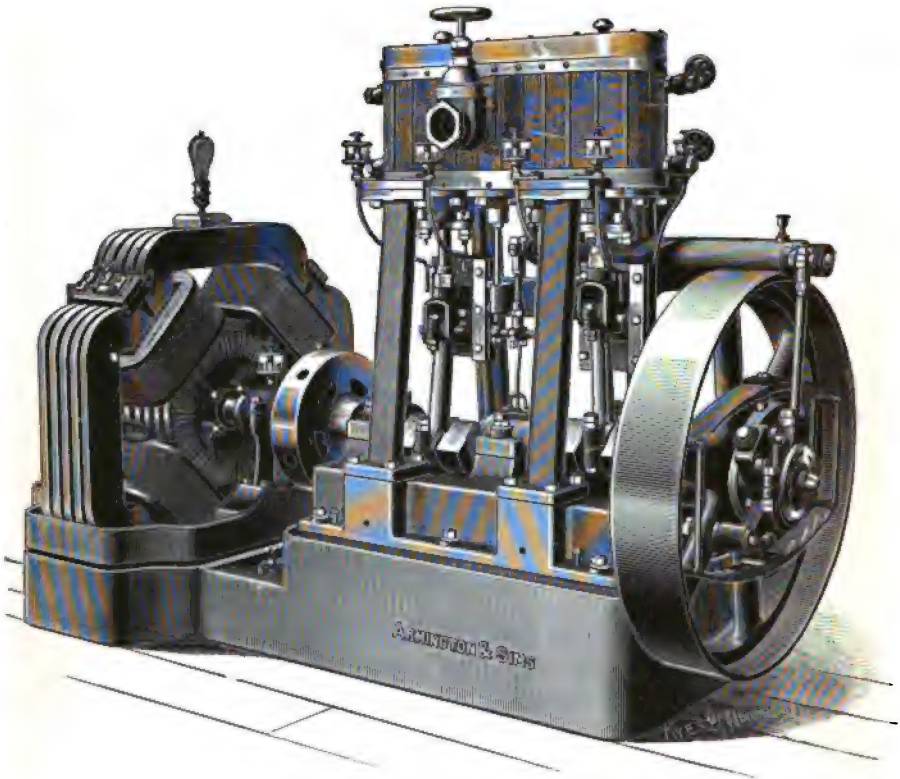

FIG. 51.—DOUBLE ENGINE FOR DYNAMOS.

its own shaft, as designed by Messrs. Armington & Sims, with opposite cranks.

The three- and four-cylinder engines of Brotherhood and others, which are completely balanced, are often coupled directly to the "dynamo" and driven at very high speeds. The Parsons turbine, making several thousand revolutions per minute, is similarly employed.

Tower's Spherical Engine, as constructed by Heenan & Froude, of Manchester, G. B., is shown in the engraving—a peculiar and very compact and fast-running machine, especially applied to the driving of electrical and other rapidly rotating

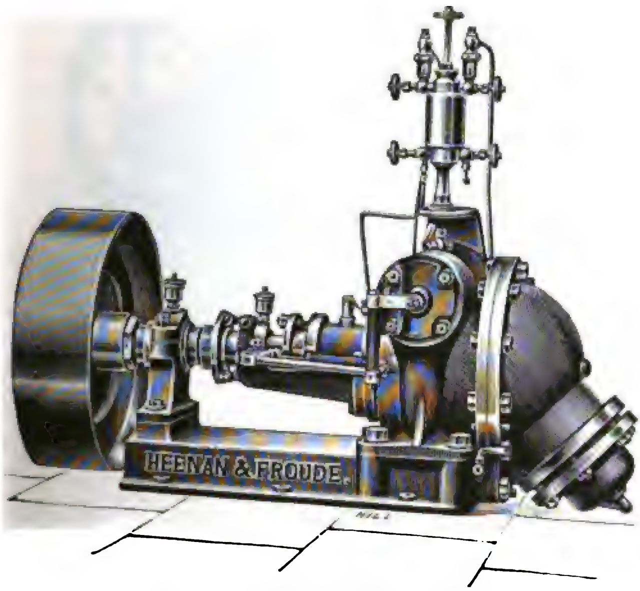

Fig. 52.—The Tower Spherical Engine.

apparatus. It consists of a spherical working "cylinder" in which a disk spins, driving its supporting shaft at any required speed. These engines are coupled direct to the dynamo, or to a fan or pump, and are much used on shipboard, when compactness and noiselessness are demanded. This is one of the most singular forms of steam-engine yet successfully introduced.

The special construction of compound engine next illustrated is that adopted by Wolf, of Magdeburg, as a standard design, which is considered to combine compactness, simplicity, small first cost, and economy, with good regulation in an unusually satisfactory manner.

They are mounted upon a hollow cast-iron frame, and their cylinders and receivers are steam-jacketed. The smaller cylinder is fitted with Rider's expansion gear, the larger with an expansion gear adjustable by hand.

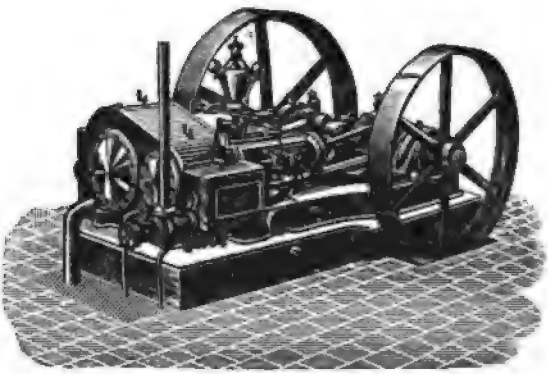

FIG. 53.—THE WOLF COMPOUND.

The following are the usual proportions, as given the Author by the designer:

Effective H. P. at 7 atmospheres (100 lbs.) pressure.	Cylinder-diameter. High Pressure. mm.	Cylinder-diameter. Low Pressure. mm.	Stroke. mm.	Revolutions per Minute.	Fly-wheel. Diameter. mm.	Fly-wheel. Width. mm.	Approx. Weight. With Condenser. kilos.	Approx. Weight. Without Condenser. kilos.
50	280	470	400	120	1880	290	9000	10600
75	350	590	460	115	2200	400	15000	17000
100	370	630	480	115	2400	400	18000	20100
125	390	660	500	115	2500	400	22500	25500

27. **Marine-engine Designs** illustrate the most remarkable applications of the inventive genius of the mechanic, the greatest talent of the designing engineer, and the best work of the engine-builder. Marine machinery must, more than any other, combine the contradictory elements of maximum power and strength, with minimum weight and cost in operation. The boiler must be compact, strong, of high power, and safe; the engine must be capable of utilizing high steam-pressure and ratios of expansion, great piston speed, high velocity

of rotation, must be compact, accessible, even when in motion in a heavy sea, free from danger of heating journals, and capable of prompt starting, stopping, and reversal. A careful study of the plans of recent engines, whether for the merchant or the naval service, will show how marvellously the engineer has succeeded in combining these qualities.*

In designing naval machinery, the fact that a usual ten-knot speed, adopted in ordinary times, either on the station or in passing from station to station, often demands but 12 or 15 per cent of the power to be supplied at full speed, must be kept in mind. In such cases it is usually advisable that the engines be designed in two or more groups which may be worked economically independently, in such manner that at minimum speed but one set may be employed efficiently; while at higher and maximum speeds two or more, or all, may be worked in conjunction. As many as four distinct, but similar, sets of multiple-cylinder engines are sometimes fitted up in naval vessels, a single one being well proportioned for low speeds; while all are required at full power. The fact that either one may be made substantially equal in efficiency to a single engine of a power equivalent to all combined is a strong argument in favor of this arrangement.

As an illustration of the general construction of the machinery of a modern U. S. war-vessel of moderate size, we have the following from a number which are designed for a cruising speed of 15 knots, and are to be propelled by machinery capable of developing, when forced, 9000 I. H. P., and under ordinary conditions 8000, driving twin screws.

The main engines are inverted, vertical, direct-acting, triple-expansion, with cylinders of 34½, 48, 75 inches in diameter and 42 inches stroke, and it is estimated that at a piston speed of 900 feet per minute, or 129 revolutions, the I. H. P. will be 9000.

Each engine and its auxiliaries is separated from the other by a fore-and-aft water-tight bulkhead, so that in case of accident to one engine the other would not be affected.

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*In this field, consult the manuals of Seaton, of Sennett, of Busley (Die Schiffsmaschine), and of Ledieu (Appareil à Vapeur de Navigation), etc.

The main valves are of the piston type, worked by Stephenson, double bar links: one valve for each high-pressure cylinder, two for each intermediate, and four for each low-pressure cylinder; the diameter of all the valves being 17 inches.

The intermediate and low-pressure cylinders are steam-jacketed at sides and bottom; the high-pressure ones have linings, but are not jacketed. The framing consists of an inverted Y-column at the back of each cylinder and two forged steel cylindrical columns at the front, as customary. The engine-bed plates will be of cast-steel supported on wrought-steel keelson plates which are built in the ship.

The piston-rods, valve-stems, and all working rods are of mild forged-steel; the pistons of cast-steel. The crank-shafts are of mild forged-steel, made in three sections which are reversible and interchangeable with each other and with those of the other engine. They are 14 inches in diameter in the journals and 15 inches in the pins, and have axial holes of 6 inches in diameter through the former and of 6½ inches through the latter.

There is a separate condenser for each engine, made with cast-brass heads and rolled-brass shell, bolted and riveted together. The castings are seven-sixteenths inch in thickness, and contain all the nozzles for steam and water openings. The shell is one-fourth inch thick, butted, strapped, riveted, and soldered together. The diameter is 5 feet 9 inches, and length between tube-sheets 10 feet 3 inches. Each condenser contains 3788 seamless drawn-brass tubes, five-eighths inch outside diameter, giving 6353 square feet of cooling-surface of the outside. Condensing-water is supplied each condenser by a centrifugal circulating pump capable of discharging 9000 gallons of water per minute from sea or bilge.

The air-pumps for each engine are two vertical, single-acting, lifting-pumps of 20 inches diameter and 18 inches stroke, driven by an engine making 2½ revolutions for each double stroke of the pump. The small spur-wheel is made of rawhide to lessen the noise when working. The cylinders of the air-pump engine are 6 inches diameter and 12 inches

stroke, and exhaust into either receiver or into the condenser at will. The pumps are entirely of "composition."

There are four main and two auxiliary boilers of mild steel, and of the horizontal, return, fire-tube type, and intended for a working pressure of 160 pounds. The main boilers are 15 feet outside diameter and 18 feet long, double-ended, and with shells $1\frac{1}{64}$ inches in thickness. Each boiler contains eight corrugated steel furnaces 3 feet internal diameter. The tubes are lap-welded or seamless drawn steel, $2\frac{1}{2}$ inches outside diameter, and the ordinary ones No. 12 B. W. G., and the stay tubes No. 6 B. W. G. in thickness. The main boilers have about 17,460 square feet of heating-surface and about 552 square feet of grate; the auxiliary boilers contain 1937 square feet of heating surface and 64 square feet of grate. The main boilers are arranged in groups of two with two smoke-pipes. The auxiliary boilers are above the protective deck. The ship is intended to cruise under very light forced or natural draught, and has been given large heating-surface for this reason.

There are a main and an auxiliary feed-pump in each fire-room, with an additional auxiliary feed-pump in each engine-room; the whole capable of supplying many times the quantity of water needed at full power. There are evaporators, distillers, auxiliary pumps, blowers for draught and ventilation, and the other auxiliaries now furnished ships of this class, together with the necessary hydraulic plant for loading and handling the battery.

The propellers are manganese bronze, with adjustable blades.

A successful modern marine engine, since it must so thoroughly combine maximum strength and power with minimum size and weights, is the most perfect known example of symmetrical and accurate proportioning in every part.

The following table is a condensed exhibit of the variety of sizes and types and of the proportions of recent vessels added to the U. S. Navy.

In recent designs by the Messrs. Thomson, a steamer intended to have a speed of 23 to 24 knots, and to cross the Atlantic in five days, is given a length of 630 feet, 70 feet beam,

VESSELS OF THE U. S. NAVY.

(All vessels have twin-screws except those otherwise noted.)

Name of Vessel.	Length between Perpendiculars.	Beam.	Mean Draught.	Displacement.	Trial Speed.	Cylinders. H.P.	Cylinders. I.P.	Cylinders. L.P.	Cylinders. Stroke.	Revolutions.	I.H.P.	Main Boilers. No.	Main Boilers. Grate-surface.	Main Boilers. Heating-surface.	Propellers. Diameter.	Propellers. Mean Pitch.	Propellers. Helicoidal Area (one screw).
	ft. in.	ft. in.	ft. in.	tons	knots*	in.	in.	in.	in.				sq. ft.	sq. ft.	ft. in.	ft. in.	cu. ft.
Chicago ... A	325	48 2¼	19 1¼	4500		45		78	57	69.3	5248.47	7	672	19950	15 6	24 6	77.93
Boston and Atlanta ... B	270 3	48 1¼	17	3189	15.5	54		2–74	42	67.2	3511.64	8	400	10146.4	17	24.235	87.1
Dolphin ... B	240	32	14.46	1485		42		78	48	74.16	2365.64	4	270	6551.03	13 9	24	69.12
Concord, Bennington, Yorktown	228	36	14	1700	16.2	22	31	50	30	161	‡3711.13	4	220	8091.88	10 6	12 6	25.42
Baltimore	315	48 9	19 6	4391.5	*20.2	42	60	94	42	116.9	10250	4	656	16674.64	14 6	20	57.168
Philadelphia	315	48 6	19 2	4325	†19.7	38	58	86	40	119.57	8815	4	624	20457.6	14 6	20 4	57.168
Newark	310	48 1¼	18 9	4088		34	52	76	40		8866	4	540	15660	14 6	19.267	57 168
San Francisco	310	48 1¼	18 9	4088	†19.518	42	60	94	36	125	9912.93	4	588	19480	13 6	18 9	57.59
Charleston ... C	300	46 2	18 6	3730	18.205	44		85	36	114.6	6666.16	6	421.2	15147	14	16 6	54.75
Vesuvius	246 3	26 5	9	970	21.64	21¼	31	2–34	20	280	4295	4	195.2	8961.18	7 9	9 4¼	15 9
Maine	310	57	21 6	6600	16	35¼	57	88	36	132	9000	8	553	18800	14 6	16 0	65.68
Texas	290	64 1	22 6	6300		36	51	78	39	123	8600	4	532	16912	14 6	17 4	66
Monterey	256	59	14.9	4000		27	41	64	30	150	5400	§6	388	12840			
Massachusetts, Oregon, Indiana	348	69 3	24	10267	16	34¼	48	75	42	128	9000	4	552	17460			
New York ... D	380	64 2¼	24	8100	20	32	48	72	42	129	16000	6	988	[illegible]1005			
No. 12 ... E	400	58	23	7350	21	42	59	92	42	129	20500	8	1221	39538.8			
No. 6	330	53	21.6	5500	20	42	59	92	42	129	13500	6	824	28298.64			
Cincinnati, Raleigh	300	42	18	3000	20	36	53	2–57	33	164	10000	4	597	19382	14	15 6	54.9
Nos. 9, 10, and 11	257	37	16	2000	17	26¼	39	63	26	185	5400	5	366	10960	11	12 5	32.1
Gunboats Nos. 5 and 6	190	32	12	1000	13	15¾	22¼	35	24	200	1600	2	100	3630			
H. D. Ram	243	43 5	15 5	2050	17	25	36	56	36	150	4800	4	288	10797			
						Qua	drupl	e 2									
Cushing	137.5	15 05	5.2	105	22.5	11¼×1	6×22¼	¼×22¼	15	370	1720	‖2	76.6	4750	4 3	8 5	8.33
Petrel ... B	176 3	31.0	11 7	870	12	25		46	33	135	1095	2	96	2796	9 9	12 6	23.55
Practice Vessel	180	32	11 6	800	12	13¼	21	31	20		1300	2	78	2640 (about)			
Torpedo Cruiser	259	27 6	9 2	800	22	23¾	34¼	2–38	18	333	6000	¶8	286	17732			
Tugboats Nos. 1, 2, and 3 ... B	92 6	20 11¼	8 0	192.4		13	20	31¼	24	100	300	1	49	1350 39			

A Beam engines.
B Single screw.
C Compound engines.
D Twin screw, 4 engines.
E Triple screw, 3 engines.
* Speed by log.
† Speed by measured distance.
‡ Concord made 3404.529 I.H.P.
§ 4 Ward's boilers, 2 Scotch boilers.
‖ Thorneycroft boilers.
¶ Tubulous boilers.

and a draught of about 25 feet. Twin-screws of 22 feet diameter are driven by triple-expansion engines of 35,000 indicated horse-power. The displacement is about 15,000 tons.

28. Ship-propulsion involves the consideration of the resistances of the vessel, the efficiency of the motive machinery, and the character and effectiveness of the instruments of propulsion. The laws of resistance with well-shaped vessels are now fairly well ascertained and formulated; and are simply those of smooth bodies exposed to the friction of fluids traversing their surfaces, the resistance of the fluid to lateral motion and displacement being in such cases comparatively small.

The useful work of the engine consists in forcing the ship forward against a resistance offered by the wind and the sea and, in vastly greater amount ordinarily, by the friction between the hull of the vessel and the surrounding water, or, more correctly, between a shell of the liquid carried along, adherent to the hull, and the water outside it. The work thus performed by the engine is expended in producing currents and eddies, which, by their own friction, are finally extinguished and the lost energy is then diffused in the mass of water in the form of heat. This resistance varies, ordinarily, as the square of the speed of the vessel, and as the area of its wetted surface. In badly formed ships it is less wholly frictional, and the resistance increases with the speed in a higher ratio.

The deduction of Dr Froude from experiment is that well-formed iron ships will offer a resistance of $0.0005 V^2$ per square foot of wetted surface; V being the speed in knots per hour. The power is approximately I. H. P. $= 0.00005 LV^3$.

In all ships having fair and well-adapted lines, the wave-making energy, for the speeds for which they are designed, may be assumed, with satisfactory accuracy, to be too small to be considered, and the resistance is practically due entirely to friction. Suppose a wave having a period due to the speed of the ship to have fitted upon it a wave-shaped solid, and the latter to move along with the wave; the frictional resistance would be that due to the area of that solid moving over the

surface of water at the assumed speed. To obtain a measure of this resistance, Rankine determined the mechanical work done while overcoming friction during an elementary period of time, *dt*, dividing that work by the distance through which the solid, as a whole, moves in that time.*

The intensity of this resistance per unit of area is the quotient of work done divided by distance traversed. By a simple process of algebraic analysis, based on this method, the expression for the total resistance is found to be

$$R = ALBV^2 (I + 4 \sin^2 \theta + \sin^4 \theta);$$

in which equation θ is the mean angle of greatest obliquity of the lines, A is a constant multiplier, B the mean wetted girth of the surface exposed to friction, L the length, and V the speed of the vessel. The *power* demanded to impel a ship is thus found to be the product of a constant to be determined by experiment, the area of the wetted surface, the *cube* of the speed, and the quantity in the parenthesis, which is known as the "coefficient of augmentation." The last term of the coefficient may be neglected in calculating the resistance of ships, as always too small, in cases to which the formulas apply, to be practically important. In applying the formula, the mean of the squares of the sines of the angles of maximum obliquity of the water-lines is to be taken for $\sin^2 \theta$, and the rule then will read thus:

To obtain the resistance of a ship of good form, in pounds, multiply the length in feet by the mean immersed girth and by the coefficient of augmentation, and then take the product of this "augmented surface," as Rankine termed it, by the square of the speed in knots, and by the proper constant coefficient selected from the following:

For clean-painted vessels, iron hulls................	0.01
For clean-coppered vessels..........................	0.009 to 0.008
For moderately rough iron vessels..................	0.011 +

* Miscellaneous Papers, p. 491; Phil. Trans., 1863; Trans. Inst. Nav. Archts., 1864; Treatise on Shipbuilding, 1864.

The result will be approximately and very closely the resistance in pounds.

The net, or effective, horse-power demanded will be quite closely obtained by multiplying the resistance calculated, as above, by the speed in knots and dividing by 326. The gross, or indicated, power is obtained by multiplying the last quantity by the reciprocal of the efficiency of the machinery and propeller, which usually should be about 0.6. Rankine uses as a divisor in this case 200 to 260.

The form of the vessel, even when designed by skilful and experienced naval architects, will often vary to such an extent as to cause the above constant coefficients to vary somewhat; and the range of variation is found to be from 0.8 to 1.5 the figures given, with good forms. The rule of the originator of the "wave-line theory" is to give the fore-body a length equal to, or exceeding, that of the after-body; and he considers that it may be a half longer than the latter with advantage. His rule for the after-body is: Take three eighths of the square of the speed in knots for its length in feet. Also, to find the best speed for a given length of after-body: Take the square root of two and two-thirds times the length in feet, and the speed is given in knots. The sum of these lengths usually is less than the proposed full length of a ship, in which case a straight middle body is introduced. The rule is apparently not exact, however, as it is found that the power demanded in the propulsion of vessels having forms quite different from, and fuller than, those produced by the wave-line, may be less than that calculated as above.

A convenient form for iron vessels of the equation just given is the following:

$$HP = \frac{SV^3}{20,000};$$

in which S is the augmented surface. Investigations made upon iron steamers of moderate size, and economical in their absorption of power, show that, though much fuller than the

wave-line theory would dictate, and although having much less easy lines, apparently, the coefficient often exceeds that given for trochoidal ships. The tests of the speed and power of the Mary Powell, a famous Hudson River steamer, having a hull of wood, coppered, and making a speed exceeding, at times, twenty miles an hour, indicated the value of the coefficient for that craft to be 23,500. The wide departure of the best river steamers from the forms of deep-sea-going craft, from which the rule adopted in calculations of resistance and power was deduced, is an illustration of the fact that the trochoidal form, or even the general proportions given by the wave-line theory, may be widely departed from, if only the lines be fine and fair, without loss of efficiency, or even sometimes with advantage. It is not the habit of the best builders to attempt the fineness of stern, far less of bow, that the Russell system would dictate as best. In fact, it is almost invariably the fact that the bow is made much shorter and fuller than the stern, and it is also the almost invariable custom, on fast as well as on slow ships, to make the after-body much shorter than the above rule would dictate.

The entrance of a pure wave water-line, according to Russell's theory, is a curve of versed sines, and the run trochoidal, the length of entrance being to the run as 3 : 2; and there should also be a fixed proportion between the speed and the

ENTRANCE AND RUN FOR A GIVEN SPEED.

Speed in Knots per hour.	Length of Entrance in feet.	Length of Run in feet.	Speed in Knots per hour.	Length of Entrance in feet.	Length of Run in feet.
1	.562	.375	11	68.00	45.38
2	2.248	1.500	12	80.93	54.00
3	5.058	3.375	13	94.98	63.38
4	8.992	6.000	14	110.15	73.50
5	14.050	9.375	15	126.45	84.38
6	20.232	13.500	16	143.87	96.00
7	27.538	18.375	17	162.42	108.38
8	35.960	24.000	18	182.09	121.50
9	45.522	30.375	19	202.88	135.38
10	56.200	37.500	20	224.80	150.00

length, which can be obtained from the table just given, or from the formulæ:

$V =$ velocity of ship in knots per hour;
$E =$ length of entrance in feet $= .562\,V^2$;
$R =$ length of run in feet $= .375\,V^2$.

There may be any length of middle body.

The following is a common nomenclature, and on the next page is given the method of computation for Rankine's method of determining resistance:

$G =$ length of mean immersed girth in feet;
$C =$ coefficient of augmentation;
$S =$ area of augmented surface in square feet;
$k =$ coefficient of propulsion;
$=$ 20,000 for a ship designed with wave-lines and iron skin;
$=$ 21,800 for a ship designed with wave-lines and copper-sheathed;
$V =$ velocity of ship in knots per hour;
$M =$ mean of squares of sines of greatest obliquity of water-lines;
$M_1 =$ mean of fourth powers of sines of greatest obliquity of water-lines;
$H =$ horse-power required to propel vessel at V speed.

$$C = 1 + 4M + M_1; \qquad k = \frac{SV^3}{H};$$

$$V = \sqrt[3]{\frac{Hk}{S}}; \qquad H = \frac{SV^3}{k}.$$

It will be seen that the form proposed by the wave-line theory is simply an expedient by which the head-resistance is reduced to an inappreciable quantity, and that the calculation of the remaining element of resistance—that due to friction—is thus made easy and accurate. But a slight reduction of length, or a slight increase in fulness of line, such as would make head-resistance perceptible, might decrease the friction, by its reduction of area of augmented surface, to a greater extent. In fact, it is certain that there exists, somewhere between the

TABLE SHOWING METHOD OF COMPUTING THE SPEED OF A VESSEL WITH A GIVEN INDICATED HORSE-POWER.

Coefficient of Augmentation.

Water-lines.	Sine of Obliquity.	Squares of Sines.	4th Power of Sines.
L. w. line.....	.370	.1369	.01874
2d w. line.....	.315	.0992	.00984
3d w. line.....	.290	.0841	.00707
4th w. line....	.265	.0702	.00492
5th w. line....	.235	.0552	.00304
6th w. line....	.165	.0272	.00074
Keel..........	.000	.0000	.00000
Means....		.0674	.00583

$1 + (4 \times .0674) + .00583 = 1.275$
= coefficient of augmentation.

Speed in Knots.

Indicated horse-power.... 5471
Coefficient of propulsion.......... 20000

Augmented surface.........36979)109420000

Cube of probable speed.............. 2959

Probable speed in knots.............14.356

Indicated Horse-power.

Cube of speed in knots........... 2959
Augmented surface.. 36979

Coefficient of propulsion.....20000)109420861

Indicated horse-power............5471.0430

Augmented Surface.

No. of Ordins.	Half-girths. Feet.	Simps. Mults.	Products.
1	21.0	1	21.0
2	27.2	4	108 8
3	30.8	2	61.6
4	34.6	4	138.4
5	38.8	2	77.6
6	41.5	4	166.0
7	42.6	2	85.2
8	44.0	4	176.0
9	44.0	2	88.0
10	44.0	4	176.0
11	43.3	2	86.6
12	42.1	4	168.4
13	40.3	2	80.6
14	38.1	4	152.4
15	36.0	2	72.0
16	35.0	4	140.0
17	32.0	1	32.0

Divide by 3)1830.6

Half No. of Int. 8) 610.2

Mean girth............. 76.3
Length of ship.......... 380

Product................ 28994
Coefficient of augment.. 1.275

Augmented surface in square feet. 36979

proportions given by the wave-line construction and those of bluff-bowed and full-sterned ships, a mean at which the sum of all resistances is a minimum. The proportions of ships are usually stated in the ratios of length to breadth; but this is only correct in cases in which the hulls compared have similar forms, as the length of bow and of after-body are, according to the accepted theory, functions of speed proposed, and not of length or breadth or size of ship. Experience, as well as approximate theoretical determinations, would indicate that,

for the common forms and sizes of vessels, a length equal to about seven breadths is not far from the best ratio for sailing-vessels, and ten to one for screw-steamers, in which latter case the location and action of the propeller is an important element in the problem. But the precise proportion which gives maximum efficiency has never yet been determined, either by experience or direct experiment.

The deductions drawn by the Author from a comparison of natural with artificial constructions for fast swimming seem to be capable of summary as follows:*

(1) As to the general principles governing the resistance of bodies moving through the water, it may be stated that the head-resistance is a minimum when the length of fore-body is such that it shall part the water with minimum disturbance of its surface, and that the wave-line theory provides a rule by which this minimum resistance may be secured; that the minimum frictional resistance is met when the surface exposed to rubbing, multiplied by the square of its velocity, is a minimum; and that the total resistance is a minimum when the sum of these two constituent quantities is a minimum. In other words, to secure least wave-making resistance, the ship should have an elongated form, with smooth lines, and such proportion of length to breadth as the wave-line theory would indicate: to obtain the least possible resistance by friction of the water on the sides of the hull, the latter must approach the sphere or hemisphere in shape; to give the lowest possible total resistance, the form must be something between these two extremes, approximating to the former or the latter more closely as the speed is greater or less; or, in other words again, the proportion of length to breadth increasing as speed increases. The higher the speed proposed, the longer the ship.

(2) The proportion of fore-body to after-body will be affected, and exactly determined, by the method of parting and of restoring the divided water. As the separation is effected by the power of the ship's engines, and the restoration is

* Forms of Fish and of Ships; Trans. Brit. Inst. Naval Archts. 1889.

caused by the force of gravity, and its rate is limited by the acceleration of gravity, it would seem from these considerations, as well as from observation of the best forms for speed, whether natural or artificial, that the after-body, for high speeds at least, should be longer than the fore-body.

(3) The character of the lines to be adopted is a matter of some uncertainty. If the requirement be simply that the water shall be parted with an accelerated motion, shall come to rest at the middle-body, and be restored to its place after the ship has passed with a regular retardation, the lines must be of the kind known as wave-lines or trochoidal lines. If the specification be that the total resistance, wave-making and frictional, shall be a minimum, the form thus designed must be shortened up, and the character of the lines may be changed. Experience and observation of natural forms indicate that the curve of displacement for the forward body should be a full line, instead of hollow, as when made on the wave-line system. The lines of the after-body are determined, in part, by the considerations above stated, and also by the action of the propelling instrument, are thus drawn out into a longer and hollower curve, and are more nearly, if not becoming actually, trochoidal. The modern screw steam-yacht, with straight bow-lines and fine run, most nearly of usual ship-forms, illustrates this ideal form.

(4) The Author finds that the best examples of naval architure, and the most perfectly formed of fishes, for fast movement through the water, have alike a proportion of length to breadth of about eight to one, and that all sizes of fish and of craft intended for speed purely have a proportion of fore-body to after-body of usually about seven to thirteen, which may be taken as indicating a strong probability that this proportion is that which makes total resistance a minimum, for all sizes, and perhaps at all speeds, when the propelling instrument is at the stern. Where there is no middle body, as in most cases in which the hull is designed for speed simply, the fore-body is 0.35 to 0.40, and the after-body 0.65 to 0.60 of the length of the ship.

(5) The forms of ship best for securing maximum speed with minimum power are evidently subject to modification, in the direction of increasing fulness, for all cases in which the problem is modified by the introduction of any commercial consideration, or the necessity of obtaining greater handiness. In even the fastest vessels such consideration must come in, in all actual cases, and the more as the question becomes more and more one of economy in other directions than that of power. It is thus that in freight traffic at moderate and low speeds the straight middle body is invariably brought in.

(6) When by the processes of evolution of forms of maximum efficiency, of adaptation of size to route and work, and of machinery of maximum power and economy of fuel with minimum weight, such as are now in progress, and doing such effective work, the steam-vessel and the sailing-craft of the world become as well fitted to their various purposes as are the forms of their prototypes among fishes, we may expect that the possibility of enormously increased speed and carrying power with greatly reduced cost in power, steam and fuel, and money will be found still more apparent than they to-day appear to the most skilful of engineers and naval architects. The forms of ships, both sailing and steaming, are probably approximating rapidly to the perfect shape that should crown such a process of development. Engineers and naval architects are gradually reducing the cost of power in steam and fuel consumed, and have already produced the horse-power on less than 1¼ pounds of coal per hour; and the weights of marine machinery have been reduced until many engineers are discussing the possibility of the application of that motor to the propulsion of balloons or other aërial machines.

According to Dr. Froude, the total resistance at the engines may be analyzed into

(1) The normal resistance of the ship as measured by the *net* screw-thrust.

(2) The added resistance due to the action of the screw itself in producing a difference of head of water forward and aft.

(3) The resistance due to friction of the screw-blades.
(4) The net friction of the machinery.
(5) The added friction due the load.
(6) The resistance of the pumps.

Resistances 1, 2, 3, and 6 may be taken as varying similarly and as the square of the speed; 4 and 5 are probably nearly constant under ordinary variations of speed of engine and of ship.

The same investigator finds that with a good disposition of machinery, as an average, the efficiency of the whole system is about 0.40; 0.60 of the indicated power being wasted, of which about 15 per cent is engine-friction.*

The obvious conditions of maximum speed are:

(1) Maximum power in a given weight and space.
(2) Minimum weight and volume of vessel.
(3) Minimum frictional and other resistance of wetted surfaces.
(4) Maximum perfection of form, having reference to the resistances to forward motion, to lateral drift, and to power—in the sailing vessel—of carrying canvas.

The General Theory of the Propeller, whether screw, jet, or paddle, is exceedingly simple; but the conditions of application are such as to make it somewhat difficult of application. Its action consists in the expenditure of energy in the backward impulsion of a mass of water, the resistance of which to acceleration supplies the force which drives the ship onward. Using Rankine's treatment and symbols,† if the "slip" of the water be s, its weight W, the resistance R, and the speed of the ship v,

$$R = \frac{Ws}{g}; \quad Rv = \frac{Wsv}{g}.$$

This impelling action must, to secure maximum efficiency of propeller, be effected by an instrument which takes hold of

* Consult White's Naval Architecture.

† London Engineer, January 11, 1867; Miscellaneous Papers, p. 544.

the fluid without shock or disturbance of the surrounding mass, and, by a steady acceleration, gives it the required final velocity of discharge. The velocity of the propeller overcoming the resistance R would then be

$$\frac{v + (v + s)}{2} = v + \frac{s}{2};$$

and the work performed would be

$$R\left(v + \frac{s}{2}\right) = \frac{Wvs}{g} + \frac{Ws^2}{2g},$$

the first of the last two terms being useful, the second the minimum lost work; the latter being the wasted energy of the water thrown backward. The efficiency is

$$E = v \div \left(v + \frac{s}{2}\right);$$

and this is the limit attainable with a perfect propelling instrument; which limit is approached the more nearly as the conditions above prescribed are the more nearly fulfilled. The efficiency of the propelling instrument is probably rarely much above 0.60, and nevér above 0.80; and the above quantity must be reduced proportionally in actual work.

The screw propeller wastes energy by sending off a mass of water in a twisting, rope-like current; all the energy of which rotation, and that of any lateral spreading as well, is wasted, in addition to that necessarily lost by the direct backward flow. Algebraic analysis cannot well be applied with satisfaction to the case; but it may be commonly safely assumed that the screw may give an efficiency of something like sixty per cent.

The mathematical discussion of the efficiency of jet propulsion is as follows:

If a is the area of the nozzle, u the velocity of the jet of water in feet with respect to orifice, and m is the mass of

one cubic foot of water, then the whole propelling force is $mau(u - v)$, where v is the velocity of the ship. This is equal to the resistance of the vessel; therefore the work done on the vessel in a unit of time $= mau(u - v)v$.

The kinetic energy given to the stream in a unit of time is $\frac{1}{2}mau(u - v)^2$, and the efficiency of the propeller is

$$E = \frac{mauv(u - v)}{mauv(u - v) + \frac{1}{2}mau(u - v)^2} = \frac{2v}{u + v}.$$

This efficiency becomes more and more nearly unity as u approaches v in value, as the water is more nearly at rest on leaving the orifice. Also as $u - v$ becomes smaller, the area of the orifice must increase for the same ship, and here, as always, the greater the quantity of water acted on by the propeller, the greater its efficiency.*

The British Government has built two jet-propulsion steamers, the Waterwitch and the Squirt. The former was a gunboat of 1100 tons, built in 1866, and tried in 1867, and attained a speed of nine knots, developing 838 horse-power and throwing 314 tons of water a minute, by means of centrifugal pumps, from two nozzles at the water line at each side of the vessel, having a combined area of six square feet. The efficiency of this apparatus was found to be 18 per cent, one half that of a similar vessel driven by a good screw. The Squirt was a small torpedo-boat, built by Thorneycroft, which developed 167 horse-power and discharged 60 tons a minute. The speed was twelve, as against seventeen knots attained by a sister-ship having a screw and equal steam-power. The efficiency was 25 per cent.

In designing the screw propeller, as was shown by Dr. Froude,† the best angle for the surface is that of 45° with the plane of the disk; but as all parts of the blade cannot be given the same angle, it should, where practicable, be so proportioned

* See Rankine in London "Engineer," January 11, 1867. Also William Kent in "Mechanics," March 1891.

† Trans. Inst. Naval Architects of G. B., 1878; vol. 19, p. 47.

that the "pitch-angle" at the centre of effort should be made 45°. The maximum possible efficiency is then, according to Froude, 77 per cent.

In order that the water shall be taken on without shock and discharged with maximum backward velocity, the screw must have an axially increasing pitch. The equation of its curve may be developed thus:

Let a = initial pitch-angle;
p_1 = initial pitch;
p_2 = terminal pitch;
c = circumference of disk;
k = fraction of circumference occupied by projection of blade.
x, y = coördinates.

With uniform increase of pitch,

$$\frac{d^2y}{dx^2} = a, \text{ const.};$$

$$\frac{dy}{dx} = ax + b.$$

When $x = 0$, $$\frac{dy}{dx} = \tan \alpha = b = \frac{p_1}{c}.$$

When $x = kc$, $$\frac{dy}{dx} = \frac{p_2}{c}$$

and

$$a = \frac{p_2 - p_1}{kc^2},$$

and hence

$$\frac{dy}{dx} = \frac{(p_2 - p_1)x}{kc^2} + \frac{p_1}{c}.$$

Integrating, and remembering that when $x = 0, y = 0$,

$$y = \frac{(p_2 - p_1)x^2}{2kc^2} + \frac{p_1 x}{c}.$$

Making the axis of X tangent to the curve at the origin,

$$y = y' + x' \sin \alpha;$$

$$x = x' \cos \alpha;$$

and taking

$$l = \sqrt{c^2 + p_1^2}; \qquad \sin \alpha = \frac{p_1}{l}; \qquad \cos \alpha = \frac{c}{l},$$

$$y + \frac{p_1 x}{l} = \frac{(p_2 - p_1)x^2c^2}{2kl^2c^2} + \frac{p_1 x}{l};$$

$$y = \frac{(p_2 - p_1)x^2}{2kl^2}.$$

Since the thrust of the screw is proportional to the area of its disk, πd^2, to the square of its motion $(Np)^2$, and as the speed of the ship is as the product Np_1 of revolutions and pitch, and the power expended, $H.P.$, is as the thrust and speed,

$$H.P. = ad^2N^2p^2 \times Np = ad^2N^3p^3;$$

and

$$d = a'\sqrt{\frac{H.P.}{N^3p^3}};$$

$$p = b'\frac{1}{N}\sqrt[3]{\frac{H.P.}{a^2}};$$

in which the constants a' and b' are given by Seaton * as

$$a' = 20,000, \quad b' = 737,$$

for fairly good ships.

The mathematical theory of the form of screws of maximum efficiency has been developed by Mr. J. N. Warrington.† Space does not permit its reproduction here.

* Marine Engineering; p. 288.

† Jour. Franklin Inst.; Sept. 1883.

The screw is commonly made of as great diameter as the draught of water will permit; but the area of the blades should be greater or less as their number is increased or diminished, being usually found best when about

$$A = a\sqrt{\frac{I.H.P.}{N}};$$

in which N is the number of revolutions per minute, and the constant a, according to Seaton, from 10 square feet for a two-bladed to 15 for a four-bladed screw. Their dimensions should be computed, evidently, as those of a cantilever. They are made as thin as possible at the edges. Their maximum breadth is usually not far from

$$b = a\sqrt[4]{\frac{I.H.P.}{N}};$$

in which a is from 15 to 20, accordingly as four or two blades are adopted. The maximum breadth should be given where the blade is at or nearest the angle 45°, and is commonly near the tip for towing-craft and nearer the hub as the speed increases and load diminishes.

The true screw is by far the more usual form of propeller, in all steamers, both merchant and naval. Twin screws, and even "triplets," are sometimes employed on the score of safety and handiness; but are not as economical, or on the whole as efficient, as the single propeller; although they have, individually, an advantage, in their better immersion, over the large single instrument.

The Paddle-wheel presents a somewhat simpler case, and some idea of the nature and extent of its wastes and their relation to the form and proportion of the wheel may be obtained by the following analysis,* thus:

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* Journal Franklin Inst., 1869; R. H. Thurston on the Paddle-wheel.

Where the radial paddle-wheel is used, a considerable part of the work done by the wheel is expended uselessly in forcing the water downward, or in lifting it; and of that power which finally acts horizontally, a part is unavoidably lost in setting in motion the water upon which the wheel acts. Thus the power exerted by the engines is considerably greater than that exerted by the ship in resisting motion.

To ascertain the amount of these losses, when the vessel is at rest, as on the instant of starting the engines, the first of

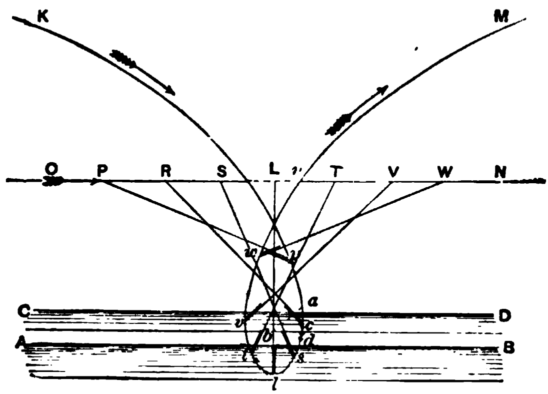

FIG. 54.—PATH OF FLOATS.

these losses, that from oblique action, is easily calculated thus:

Measure the length of the immersed arc described by the centre of pressure of the floats, and the length of its chord. The ratio of the squares of these quantities will give the ratio of total power exerted to power expended horizontally.

When the vessel moves ahead, the conditions of the problem are at once greatly changed. The floats, instead of moving in a circle through the water, describe the curve known as the cycloid. They then move through the water obliquely, and with velocities varying every instant. This forward movement gives, to some extent, a "feathering" action to the floats, and, with a properly proportioned wheel, the loss from oblique action becomes very greatly diminished, as the principal por-

tion of the work is done in the lower portion of the path of the float, where there is less oblique action.

The amount of "dip" influences the amount of loss from oblique action very greatly when the vessel moves ahead, as well as in the previous case, and the most economical wheel will be that in which the sum of the losses from dip and slip are reduced to the greatest extent.

To ascertain the amount of loss of power in oblique action:

Let V = velocity of paddle-float at centre of pressure;
v = velocity of vessel;
α = angle included between the arm and horizontal line through the centre, at a given instant;
β = the angle denoting any other position;
$\mu = \frac{v}{V}$; r = radius.

The tangential velocity of the centre of pressure of any float will be $(V - v \sin \alpha) = V(1 - \mu \sin \alpha)$, and the horizontal velocity through the water will be $(V \sin \alpha - v) = V(\sin \alpha - \mu)$.

The normal pressure on a unit of area of the float will be measured by the square of its velocity in the direction of that pressure, or $V^2(1 - \mu \sin \alpha)^2$, and the total work done by the engine in turning the float through an arc $d\alpha$ will be

$$rV^2 \int_\alpha^\beta (1 - \mu \sin \alpha)^2 d\alpha. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The horizontal component of the pressure on the float, which only is useful in propelling the ship, will be

$$V^2(1 - \mu \sin \alpha)^2 \sin \alpha.$$

While the float is moving through the arc $rd\alpha$ in the time $dt = \frac{rd\alpha}{V}$, this horizontal resistance will be met through a space $V(\sin \alpha - \mu)dt = rd\alpha(\sin \alpha - \mu)$, and the power expended horizontally upon the water while the wheel moves through any arc will, therefore, be expressed by

$$rV^2(1 - \mu \sin \alpha)^2 \sin \alpha(\sin \alpha - \mu)d\alpha. \quad . \quad . \quad . \quad (2)$$

But while this amount of work is expended upon the water, the wheel must, in consequence of the slip, do a greater amount of work in a proportion varying at each point with the slip, and the total horizontal work of the wheel will be obtained by multiplying the expression (2) by the factor $\frac{V}{V(1-\mu\sin\alpha)}$.

The total amount of work done in this direction, then, while passing through the arc, will be, including slip,

$$\gamma V^2\int_\alpha^\beta (1-\mu\sin\alpha)(\sin\alpha-\mu)\sin\alpha d\alpha. \quad . \quad . \quad (3)$$

The ratio of (3) to (1) will be the ratio of gross horizontal work to the total work done on the wheel.

Integrating between the limits, β and α, this ratio becomes

$$\frac{\frac{\mu^2+1}{2}(\beta-\alpha)+2\mu(\cos\beta-\cos\alpha)-\frac{\mu}{3}(\cos^3\beta-\cos^3\alpha)-\frac{\mu^2+1}{4}(\sin 2\beta-\sin 2\alpha)}{\frac{\mu^2+2}{2}(\beta-\alpha)+2\mu(\cos\beta-\cos\alpha)-\frac{\mu^2}{4}(\sin 2\beta-\sin 2\alpha)}.$$

In this equation, making α the angle of entrance of the mean centre of pressure of the float and calling $\beta=90°$, $\frac{v}{V}$ being known, the loss of power in any wheel by oblique action may be obtained by subtracting the ratio resulting from our final equation from unity.

The following are data assumed and results obtained as above:

First Case.	Second Case.	Third Case.	Fourth Case.
$\alpha=60°=\frac{\pi}{3}=1.047$	$\alpha=42°=\frac{7}{30}\pi$	$\alpha=30°=\frac{\pi}{6}$	$\alpha=0°=0$
$\beta=90°=\frac{\pi}{2}=1.570$	$\beta=90°=\frac{\pi}{2}$	$\beta=90°=\frac{\pi}{2}$	$\beta=90°=\frac{\pi}{2}$
$\frac{v}{V}=\frac{3}{4}$	$\frac{v}{V}=\frac{3}{4}$	$\frac{v}{V}=\frac{3}{4}$	$\frac{v}{V}=\frac{3}{4}$
Dip. = $\frac{3}{16}$ radius	Dip. = $\frac{1}{3}$ radius	Dip. = $\frac{1}{2}$ radius	Dip. = to centre
Efficiency = .949	Efficiency = .907	Efficiency = .83	Efficiency = .70
Loss by oblique action = .051	Loss = .093	Loss = .17	Loss = .30

The first case corresponds to that of light river steamboats, the second to that of ocean steamers, while the third is an exaggerated, though not remarkably rare, case. The fourth case is given simply in illustration of the use of the formula ; it, of course, is unknown in practice.

Here the efficiency and loss are expressed, not in fractions of total indicated horse-power, but of the power applied to the wheel, and, referred to I. H. P., those losses would be .047, .086, .158, and .28, respectively. If it is required to find the

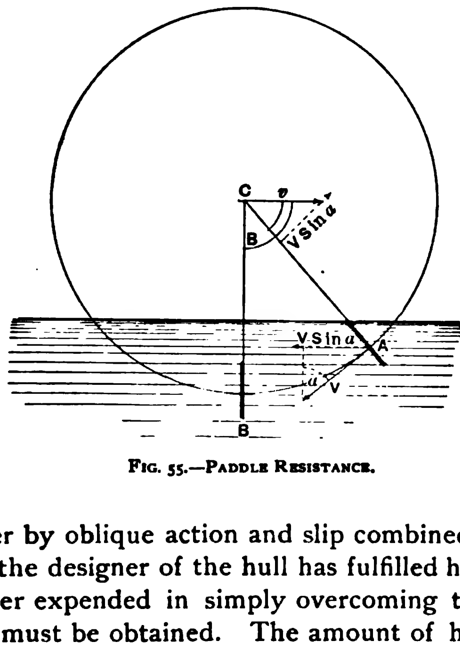

FIG. 55.—PADDLE RESISTANCE.

loss of power by oblique action and slip combined, and thus to ascertain if the designer of the hull has fulfilled his guarantees, the net power expended in simply overcoming the resistance of the ship must be obtained. The amount of horizontal expenditure of power just obtained must, in this case, be reduced by multiplying by the ratio $\frac{V - v}{V}$, or it may be taken from (2).

In the examples given above, the slip is assumed to be known, and in each case is taken at 25 per cent, and this net power usefully expended is found, after making this reduction,

to be .714, .685, .631, and .52 of total I. H. P. In the fourth case, the slip would actually be very much greater than is assumed above, and the efficiency would probably fall below twenty per centum of the indicated horse-power.

The paddle-wheel is still used in shoal waters; but it has elsewhere been, long since, almost entirely superseded by the screw propeller; which latter instrument is not only more economical in its expenditure of power, but also permits the use of vastly lighter, as well as more efficient, forms of engine. The paddle-wheel is particularly objectionable and dangerous at sea, where the rolling and pitching of the ship introduces strains of great severity, and causes such variations of speed, such "racing" of the engines, as is certain to produce danger to the machinery.

The older radial form of wheel, unless made of very great size, loses, as has been seen, a large proportion of its efficiency through oblique action. In order to evade this loss, "feathering" paddles are used, in which, by a self-adjusting arrangement, the floats of the wheel are made to take a similar series of positions in a small wheel to those obtained by the larger radial paddle-wheel. This form of wheel is commonly given one half the diameter of the corresponding radial wheel, and then a heavy "paddle-box" is avoided, with all its attendant disadvantages. The greater complication of the feathering wheel renders it peculiarly liable to derangement, and it should be designed with great care and very strongly built, particularly if to be employed where ice is likely to be met with. Their floats are sometimes of steel, and all bearing surfaces are usually of brass or bronze.

The most economical speed to run in a tideway, or against a stream, is half as fast again as the stream; for—

Let $x =$ the speed of the ship;
$v =$ the velocity of the tide or current;
$x - v =$ the progress made by the ship.

The consumption of fuel varies as the cube of the speed, x^3.

Let the consumption be cx^3; then we have the consumption for each mile to be the most economical consumption.

$$P = \frac{cx^3}{x - v} = \text{minimum}.$$

Then

$$\frac{3cx^2(x - v)dx - cx^3dx}{(x - v)^2} = 0;$$

$$3cx^2(x - v) - cx^3 = 0;$$

$$3(x - v) - x = 0;$$

$$x = \frac{3v}{2}.$$

The Power required to drive the vessel is easily computed, once the "augmented surface" is known. This gives the resistance at the assumed speed; which being multiplied by the speed of the centre of the paddle-float relatively to the ship—speed of vessel plus slip—will give the net dynamometric power of the machinery. To this must be added friction and wastes of wheel, shafting, and mechanism, usually amounting to 25 per cent or more, to obtain the required indicated power. The following are figures given by Rankine, from experience with feathering wheels. It will be remembered that "radial" wheels must be usually given about a double diameter to secure equal efficiency.

ACTION OF FEATHERING WHEELS.

Ratio of Slip to Speed.		Efficiency	Ratio A. S. to
Slip.	Paddles.	of Wheels.	Area of One Float.
0.10	0.09	0.91	124
0.15	0.13	0.87	196
0.20	0.17	0.83	272
0.25	0.20	0.80	354
0.30	0.23	0.77	442
0.40	0.29	0.71	634
0.50	0.33	0.67	850

The Distribution of Power is taken by Mr. Haswell, for cases of good design, thus: *

SIDE-WHEEL.	Per Cent.		SCREW.
Engine friction	13.83	18.83	
Oblique action	18.00	6.83	Friction of screw.
Slip of wheels	15.37	26.20	Slip " "
Utilized	52.80	48.14	Utilized.
	100.00	100.00	

The weights of marine engines, according to the same authority, are ordinarily not far from those given below:

WEIGHTS OF MARINE ENGINE.

Engine.	No. Cyls.	Vol. cu. ft.	Wt. per cu. ft.	Service.
Beam (Side-wheel),	1	63	1100 lbs.	River.
" " "	1	430	1225	Sea.
" " "	2	216	1040	"
" " "	2	253	1480†	"
Vertical (Screw),	1	4	8535†	"
" "	1	12.5	7280†	"
" "	4	506	4958†	"
" Comp.	2	4.8	10,641†	Coast.
" "	2	24.3	7500†	Sea.
" "	2	425	4380†	"

Summarizing the conclusions of Dr. Froude, who has made the most satisfactory researches in this field: ‡

A = area of propelling plane surface in square feet;

P = normal pressure on a plane area moving on a path forming the angle θ with the plane, in lbs.;

p = coefficient of pressure, lbs. per square foot = about 1.7;

f = coefficient of skin friction per square foot = about .008 (being .004 for each surface);

$k = \frac{f}{p} = .0047$ approximately;

F = friction on surface of plane moving edgeways $= Afv^2$;

* Pocket Book; p. 662.

† With boilers.

‡ Trans. Inst. Naval Architects, 1874-8.

V = speed of propeller plane, feet per second;
v = speed of vessel in feet per second;
v_1 = speed of plane through water, feet per second;
s = speed of slip;
θ = slip angle $= \sqrt{k}$ for maximum efficiency;
α = virtual pitch angle,
ϕ = actual " " $= \alpha + \theta$;
R = propulsive force to maintain speed of ship, lbs.;
W = gross work done, foot-lbs.;
w = effective " " " $= Rv = RV \tan \alpha$;
E = efficiency $= W \div w =$ about .77 as a maximum.

$$\text{Slip ratio} = \frac{s}{v+s};\ P = pAv^2 \sin \theta;$$

$$W = pAV^2 \sec^2 \alpha (\sin \phi \sin \theta + k \cos \alpha);$$

$$A = \frac{R}{pv^2 \operatorname{cosec}^2 \alpha (\cos \phi \sin \theta - k \sin \alpha)}.$$

Superiority is observed in deep rather than in broad vessels. It is computed that the power is absorbed nearly as follows:

Ship's net resistance	.426
Augmentation by negative pressure caused by screw	.170
Water friction of screw	.043
Friction of engine	.143
Friction of working load	.143
Friction of air-pump and feed	.075
Total	1.000

The proportion of gross indicated horse-power to effective horse-power is as 2.347 to 1; and if to the gross indicated horse-power be added 10 per cent for slip, the proportion will be 2.58, which nearly agrees with experiment.

At moderate speeds the resistance coincides with the calculated resistance due to surface friction alone; but as the speed increases, the entire resistance becomes more and more in excess of that due to surface friction.

In these calculations the motion of water in the ship s wake has been disregarded.

The area which will drive a ship with a given "slip ratio" is directly as the vessel's resistance, and inversely as the square of the speed; and since at moderate speeds a ship's resistance is proportional to the square of the speed, the same area of propeller will at moderate speeds drive a given ship with the same slip ratio; and areas directly as the squares of the respective dimensions of two similar ships will drive with the same slip ratio, since the wetted surface measures the resistance in each case. At the higher speeds the slip ratio will increase with the given propelling area.

The maximum of efficiency is not produced by extending the area of the propelling plane so as to minimize the slip, but the slip angle that gives the maximum efficiency is moderate. If friction did not exist, the obliquity with which the propeller acts on the water would cause no loss in efficiency. The value of θ, which gives the maximum efficiency, is the same whatever be the value of ϕ. Although the slip angle ought to have the same value whatever be the pitch angle, the slip *ratio* will be greater for large pitch angles than for small. If the slip angle be that which gives the maximum efficiency, then to produce the maximum efficiency the propelling plane ought to stand at an angle of 45° with the line of ship's motion, whatever be the coefficient of surface friction or of normal pressure. If the slip angle exceed that which gives the maximum efficiency, the pitch angle must also be increased; if the excess be small, the pitch angle must be increased by the same amount; if the excess be large, the increment of pitch angle must be still greater.

The calculations point to the conclusion that a very much longer pitch than has commonly been adopted is favorable to efficiency; and that, instead of its being correct to regard a large amount of slip as a proof of waste of power, the opposite conclusion is the true one.*

* Molesworth.

From the same source we obtain the following data as representative of good practice: *

The following figures may be taken as a good approximation of the consumption per I. H. P. per hour when the engines are being driven at a moderate speed:

	Compound Engine.	Expansive Engine.
Above 2000 I. H. P.	2 lbs.	$3\frac{1}{4}$ lbs.
Between 1000 and 2000 I. H. P. .	$2\frac{1}{4}$ to $2\frac{1}{2}$ lbs.	4 to $4\frac{1}{2}$ lbs.
Under 1000 I. H. P.	$2\frac{1}{2}$ to 3 lbs.	$4\frac{1}{2}$ to 5 lbs.

WEIGHT IN CWTS. PER I. H. P.—PROCTOR.

I. H. P.	Engines.	Boilers.	Screw Shafting.	Spare Gear.	Extra Work.	Total.
9000 to 5000	1.0 to 1 2	1.2 to 1.5	.25 to .28	.125 to .13	.13 to .15	2.705 to 3.26
5000 to 1000	1.2 to 1.3	1.5 to 1.9	.28 to .29	.13 to .20	.15 to .06	3.26 to 3.75
1000 to 500	1.3	1.9 to 2.8	.29	.20 to .29	.06 to .04	3.75 to 4.75

The above weights are for simple engines; compound engines average from 10 to 20 per cent heavier.

The general dimensions and weights of marine engines of the recently standard types are well illustrated in the next table (pp. 192, 193).

29. **Locomotive Engine designing** includes, in addition to the proportioning of engine and boiler, proper, the adaptation of the machine to its special purposes. It must, like the marine engine, be both light and powerful; but it differs from that type in the fact that it cannot make use of a vacuum, and also because the cost of fuel is not usually a matter of such supreme importance. It must also be made portable; the boiler must serve as a frame or foundation; and the machine, as a whole, must be so proportioned that it may have as nearly as practicable the needed pulling power without either unnecessary

* For an elaborate memorandum of dimensions of compound marine engines and their proportions, by Sir Fred. Bramwell, see Trans. Inst. Mech. Engrs.; 1872.

PARTICULARS OF MARINE SCREW ENGINES, BY MESSRS. MAUDSLAY, SONS, AND FIELD, 1890.

Particulars.	Agincourt, A. Common Engine.	Mesoudihye, F. Common Engine.	König Wilhelm, F. 3-cyl. Engine.	Lord Warden, A. 3-cyl. Engine.	Prince Consort, A. Common Engine.	Swiftsure. A. Common Engine.	Adriatic, M. Compound Engine.	Penelope, A.⊙ 3 cyl. Engine.
Nominal horse-power. Collective	1350	1250	1150	1000	1000	800	650	600
Indicated " "	6867	7400	8344	6705	4234	4913	3666	4702
Length of stroke in ft. and ins.	4 6	4 0	4 6	4 6	4 0	4 0	5 0	2 6
No. of revolutions per minute	61.5	66	64	63.3	56.5	68.3	52.0	103
" " cylinders	Two.	Two.	Three.	Three.	Two.	Two.	Four.	Six.
Diameter of cylinders in ins.	101	116	95	91	92	98	{ 2–42 / 2–80 }	55
" " propeller in ft. and ins.	24 6	23 0	23 0	23 0	21 0	20 0	22 0	14 0
Pitch " " " "	26 6	19 6	24 0	25 0	22 6	22 0	30 3	15 6
Diameter " " shaft in ins.	20.0	20.5	19.5	19 0	18.0	18.0	17.5	11.5
Weight of boilers in tons	250.0	228.0	220.0	195 0	183.0	136.35	226.0	116.7
" " water in boilers in tons	195.0	174.0	164.0	154.0	148.0	105.0	153.0	92 0
Total weight of machinery and water in tons	1009.0	1067.0	2200	997.0	796.0	663.0	876 0	578.0
No. of boilers	Ten.	Nine.	Eight.	Nine.	Eight.	Six.	Twelve.	Four.
Length of boilers in ft. and ins.	14 6	{ 7–18 0 / 2– 9 6	17 10	{ 7–14 6 / 2– 7 6	4–18 0 / 4–11 0	4–18 10 / 2–7 8 }	10 2	19 8
Breadth " " "	11 8	11 4	12 2	12 0	12 4	11 4	8 5	12 4
Height " " "	12 8	12 0	12 4	12 10	12 10	11 10	14 3	12 0
Steam-pressure in cylinders in lbs. per sq. in.	25.55	21.8	22.4	19.9	23.25	—	—	20.9
" " boilers " "	25	30	30	25	25	30	60	30
No. of funnels	Two.	Two.	Two.	Two.	Two.	One.	One.	One.
Diameter of funnels in ft. and ins.	{ 1–9 0 / 1–7 5	8 2	8 0	7 3	{ 1–8 3 / 1–6 4	9 0	9 10	7 9
Height of funnels above top of boiler in ft. and ins.	54 6	62 0	58 0	50 0	51 0	52 6	54 10	57 0
Total area of fire-grates in sq. ft.	951	900	900	704	704	570	494	433
" " " heating-surface in sq. ft.	27,180	22,500	22,600	20,230	22,100	15,280	14 480	11,880

Particulars.	Celt, M. Compound Engine.	Pleiades, M. Compound Engine.	Roman, M. Compound Engine.	A. Apear. M. Comp. Engine.	Timor, M. Compound Engine.	Kornilof, M. Common Engine.	Ringdove, A. ⊙ Common Engine.	Oleg, M. Compound Engine.
Nominal horse-power. Collective	250	235	230	215	215	170	160	160
Indicated " "	1215	1160	1041	1258	1234	992	956	900
Length of stroke in ft. and ins.	4 0	4 0	3 9	3 9	3 9	3 6	1 6	3 0
No. of revolutions per minute	62	56	67	62	66	60	120	60
" " cylinders	Two.	Two.	Two.	Two.	Two.	Two.	Four.	Two.
Diameter of cylinders in ins.	1–36, 1–72	1–30, 1–70	1–35, 1–70	1–36, 1–68	1–36, 1–68	52	32	1–32, 1–62
" " propeller in ft. and ins.	16 0	16 0	14 0	16 0	16 0	16 0	8 6	12 0
Pitch " " " "	20 0	21 0	19 0	19 0	19 0	18 0	12 9	18 6
Diameter " " shaft in ins.	12.5	13.5	12.0	11.5	11.5	11.0	6.25	10.25
Weight of boilers in tons	65 0	86.0	63.0	56.8	52.0	38.5	38.4	40.0
" " water in boilers in tons	45.0	58.0	43.8	36.0	35.4	25.5	24.0	30.9
Total weight of machinery and water in tons	278.0	318.0	—	237.0	232.0	178.0	138.0	172.0
No. of boilers	Four.	Two.	Four.	Two.	Two.	Two.	Four.	Two.
Length of boilers in ft. and ins	11 8	24 6	11 2	10 9	17 0	11 9	2–109, 2–90	16 6
Breadth " " "	9 2	10 3	9 0	16 5	10 3	11 4	9 8	9 6
Height " " "	11 8	12 9	11 2	10 9	10 3	12 6	7 0	9 6
Steam-pressure in cylinders in lbs. per sq. in.	—	—	—	—	—	18.1	27	—
" " boilers " "	64	76	60	78	60	30	30	65
No. of funnels	One.	One.	One.	One.	One.	One.	Two.	One.
Diameter of funnels in ft. and ins.	5 10	7 0	5 6	5 8	5 8	4 9	4 7	4 5
Height of funnels from top of boiler in ft. and ins.	51 0	45 0	41 0	47 0	48 0	38 0	33 0	37 6
Total area of fire-grates in sq. ft.	150	192	140	149	128	130	112	120
" " " heating-surface in sq. ft.	4820	4570	4412	4560	4180	3440	3200	2920

A. British Admiralty. F. Foreign service. M. Mercantile marine. ⊙ Twin screws.

mass, on the one side, or, on the other, such deficiency of weight on the driving-wheels as may produce insufficient adhesion to the track to bring into use its whole powers of traction.

In the details of its construction some special points demand attention. Its steaming power must be, in some approximate degree, proportional to its load and speed. The blast of the exhaust must be adjusted to this requirement; but if it is too "sharp," the back-pressure in the cylinders becomes so great as to reduce the power of the engine and to decrease its economy in the use of steam. The experience of years, only, could bring about the best reconciliation of so many, often conflicting, conditions; and the result has been the production of the standard types of engine now familiar to every one and partly described in Part I. The general construction of the steam-engine of this class is coming to be substantially the same all over the world, and characteristically typical. It has always a high-pressure multitubular boiler, carrying a direct-acting engine of the simplest construction, having a reversible link-motion valve-gear, a "steam-blast," and well-settled general proportions.*

For example, the following are the proportions adopted for standard engines by experienced builders;‡ the diameter of the cylinder being the unit:

Area of steam-ports..............	$= D^2 \times .08$
Area of eduction-port............	$= D^2 \times .18$
Diameter of piston-rod...........	$= \frac{D}{7}$; $D \times .16$
Diameter of feed-pump plunger....	$= D \times .12$ if of the same stroke as the engine.
Diameter of feed-pipe............	$= D \times .12$
Diameter of valve-spindle.........	$= D \times .09$
Diameter of outside crank-pin.....	$= D \times .26$
Length of " "	$= D \times .28$
Diameter of boiler...............	$= D \times 3.11$

* Consult Forney's Catechism of the Locomotive, and D. K. Clark's works.

† Molesworth.

Diameter of steam-pipe.......... $= D^2 \times .02$
Diameter of blast-pipe........... $= D \times .3$
Diameter of piston-rod........... $= D \times .16$
Thickness of piston .... $= D \times .28$
Diameter of connecting-rod ends.. $= D \times .16$
" " " middle $= D \times .21$
Diameter of crank-axle........... $= D \times .4$
Length of journal of crank-axle.... $= D \times .43$
Capacity of tenders in gallons..... $= D \times 90$ $D \times 150$
Capacity of tanks of tank-engines,
in gallons.................... $= D \times 60$
Heating-surface................. $= 65$ area of fire-grate.

$T =$ travel of slide in inches;
$L =$ lap of slide in inches;
$l =$ lead of slide in inches;
$L = T \times 0.22$;
$l = T \times 0.07$.

In locomotive design, as in marine engineering, it is desirable to secure as light parts as is consistent with the needed adhesion. It is especially important to make the reciprocating parts, the counterbalancing, and all pieces subjected to the jar of the rolling system, uncushioned by springs, just as light as is consistent with strength. They should be made of the strongest and toughest available material, and designed with exceedingly great care, thus enabling the production of flat places on the rail, or breakage, to be made less likely.

In designing compound locomotive engines, a proportion of 2 to 2.5 to 1 of low-pressure to high-pressure cylinder-volumes has usually given, on the whole, best results; later constructors favoring the higher ratio with increasing pressures. It is usually thought advantageous to make, if practicable, both ratios of expansion variable. They should always be so designed that full power of haulage may be obtained at starting. It is obvious that this system is especially adapted to engines for high speed and light loads. The double-ported valve is preferred, as giving a less "wire-drawing." The re-

ceiver-capacity is best made quite large, and should not be less than that of the small cylinder, as it must at least contain one charge of steam. While, in ordinary operation, it is advisable that the work should be as equally divided as possible, the gear should be so constructed as to permit maximum power to be secured, when needed, by following nearly full-stroke in the high-pressure cylinder. Special care should be taken to provide efficient drainage from the cylinders. It is further evident that nearly equal ratios of expansion in the two cylinders are probably best.

The valves and proportions of the compound engine must usually be so designed as to enable the compression to be kept down in spite of the very high back-pressure on the small cylinder. This compels large clearances. Similarly, the low initial pressure in the large cylinder produces the same exigency. Mr. Von Borries, an engineer of large experience, finds the following good proportions:

Cylinder.	Width port.	Travel.	Lap.	Lead.
H.P.	1.75	5.25	$1\frac{1}{8}$	$\frac{3}{16}$
L.P.	2.00	"	$\frac{15}{16}$	"
H.P.	1.75	"	$1\frac{1}{8}$	"
L.P.	2.00	"	"	"
H.P.	1.60	3.00	1.4	0.1
L.P.	2.00	"	"	"
H.P.	1.35	2.65	1.3	"
L.P.	1.44	2.58	1.15	.06
H.P.	1.00	4.20	1.12	.02
L.P.	1.16	4.50	"	.03

Every case should, however, be carefully laid down before finally deciding on the proportions and dimensions.

"*De Pambour's Problem*" finds its best illustration in the design of the locomotive engine. If, in any engine, we have the equation

$$\frac{R}{A} = \frac{U}{v_2 - v_1'};$$

in which the first member is the total resistance, R, per unit of area of piston, A; and the second is the mean effective pressure of the fluid, the quotient of the work done, U, by the volume traversed by the piston, per stroke; the principle expressed may be stated as follows:

*In a heat-engine moving with uniform periodical motion, the mean effective pressure of the fluid is equal to the total resistance per unit of area of piston.**

The proper mode of applying this principle to the steam-engine may be summed up as follows:

The resistance, R, is known from data independent of the action of the steam.

The resistance, being fixed, fixes the mean effective pressure; and the action of the fluid *adjusts itself* until the pressure balances the resistance: if the mean pressure is at first greater than the resistance, the motion of the engine is accelerated, and the pressure is diminished until the mean effective pressure balances the resistance. If the mean pressure is at first less than the resistance, the engine is retarded until the same adjustment is effected. Then the number of strokes in a given time can be computed.

In De Pambour's solution the *weight of steam* produced was treated as a constant quantity; while in Rankine's work the *available heat* of the furnace is treated as a known constant quantity; *the problem being, when that quantity, and the useful resistance to be overcome by the engine, and the back-pressure, and also the ratio of expansion, are given, to find the mean velocity with which the piston will move.*†

Let R_1 be the useful resistance at the piston. Then the total resistance is

$$R = (1 + f)R_1 + R_0.\ddagger \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Divide this by the area of the piston, in a single-cylinder en-

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* Rankine; Steam-engine, p. 341.

† Ibid., p. 424.

‡ In which, often, $f = 0$. See Part I, §§ 132–4, pp. 540–570.

gine, or by the area of the larger piston in a multiple-cylinder or compound engine, and the mean effective pressure,

$$p_m = R \div A. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If r and r' are the real and the apparent ratios of expansion,

$$r = \frac{r'(c + 1)}{1 + cr'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and compression waste being neglected or made zero,

$$p_e = \frac{R}{A} \cdot \frac{1 + c}{1 + cr'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$p_m = p_e - p_3, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

when the back-pressure is p_3. Next computing $p_1 \div p_m$, the initial pressure can be computed, and the engine will adjust its own speed and resistances to produce the value of p_1 ascertained by multiplying (5) by $\frac{p_1}{p_m}$.

Taking, as previously, H_1 as the value of the heat of the steam used, and rv_1 as measuring the volume of steam per stroke, E the efficiency of fluid,

$$p_1 \div E = H_1 \div rv_1 = p_k. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The volume swept through per minute will be

$$NAL = r'wh \div rp_k, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and we thus obtain for the number of revolutions

$$N = \frac{r'}{r} \frac{Lhw}{Ap_k}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

wh being the heat produced by the combustion of a given weight, w, of fuel, in the unit of time.

Thus, for example, assume, with Rankine,—

Data:

Resistance of train........................	10,000 lbs.;
Circumference of wheel	18 feet
Stroke of piston	2 "
Joint area of the two pistons..............	618 inches;
p_f, friction resistance per sq. in............	2 lbs.;
Back-pressure, per sq. in...................	4 "
Coal burned per minute.....................	30 "
Available heat per lb......................	6,000,000 ft.-lbs.

Results:

$$R = 10,000 \times \frac{19}{4} = 47,500 \text{ lbs.};$$

$$\frac{R}{A} = \frac{47,500}{618} = 76 \text{ lbs. per sq. in.};$$

$$p_m - p_3 = 76 + 1 - 4 = 73 \text{ lbs. per sq. in.};$$

$$\frac{p_1}{p_m} = \left(\text{assume } \frac{10}{9}\right);$$

$$p_1 = \frac{10}{9} \times 73 = 81 \text{ lbs.};$$

$$p_k = 15\tfrac{1}{2} \times 81 = 1256, \text{ approximately};$$

$$Ap_k = 776,208 \text{ lbs.};$$

$$wh = 6,000,000 \times 120 = 720,000,000;$$

$$V = \frac{wh}{Ap_k} = \frac{720,000,000}{776,208} = 920 \text{ ft. per m.};$$

$$N = V \div 4 = 230 \text{ per minute.}$$

The centre of gravity of the counterbalance weight should be in the prolongation of the line that bisects the angle formed by the two cranks, or at an angle of 135° with each crank.

The following, which is Rankine's formula, can only give approximate results, as there are other disturbing influences. The engine should be further adjusted by suspending it by means of chains or ropes from the corners of the framing, leaving it free to oscillate, and the weights should be increased or diminished until the oscillation of the engine working at speed is reduced to a minimum. The suspension should be on springs, so as to allow vertical as well as horizontal motion. The balance is sometimes facilitated by placing the driving-wheels in the lathe-centres, so that they may revolve freely, and then attaching in position the connecting-rod and other moving parts. *

Let W = accumulated weight of moving parts acting on crank;
w = weight of counterbalance;
R = radius of centre of crank-pin;
r = radius of centre of gravity of counterbalance weight;
D = distance of centre of crank from centre of engine;
d = distance of the centre of gravity of counterbalance weight from centre of engine.

$$w = \frac{WR}{r}\sqrt{\frac{D^2 + d^2}{2d^2}}.$$

The hauling-power of the locomotive is measured by the product of the weight on the driving-wheels into a coefficient of friction which depends on the condition of the wheels and rails, varying from 0.15 to 0.25 when clean and dry; but when damp and "greasy," 0.07 or 0.05.

The proportion of the weight of the engine which rests on the driving-wheels may be estimated at from one third to unity. One half is the proportion in six-wheeled engines with one pair of driving-wheels; two thirds, in four coupled wheels; and with all the wheels coupled, as used for heavy trains, the whole weight rests on driving-wheels.

* Molesworth.

The weights of locomotive engines range from 5 to 100 tons; ordinary weights are from 20 to 40 tons.

The effect of the blast-pipe depends upon its diameter and position, on the size of the chimney, and on the dimensions and proportions of the fire-box, the tubes, and the front connection or smoke-box. Mr. D. K. Clark has shown that the vacuum in the smoke-box is about 0.7 of the blast-pressure, and in the fire-box ranges from $\frac{1}{3}$ to $\frac{1}{2}$ of that in the smoke-box; that the evaporation varies nearly as the square root of the vacuum in the smoke-box; that the best proportions are those which enable a given draught to be produced with minimum back-pressure; and that the following proportions are best:

Sectional area of tubes within ferules = $\frac{1}{5}$ G. S.
Sectional area of chimney.......... = $\frac{1}{15}$ G. S.
Area of blast orifice................ = $\frac{1}{65}$ G. S.
Length of chimney................ = its diameter × 4.

If the tubes are smaller, the blast orifice must be made smaller.*

In locomotive engines Mr. D. K. Clark finds that the *excess* of the back-pressure above the atmospheric pressure varies nearly—

As the square of the speed;

As the pressure of the steam at the instant of the commencement of the exhaust;

Inversely as the square of the area of the orifice of the blast-pipe.

Mr. Clark also finds that the back-pressure is less the greater the ratio of expansion; and that it is increased by the presence of water, being greater in unprotected than in protected cylinders in the ratio of 1.72 to 1.

"With a mean of 16 per cent of release, with an admission of half stroke, and with a speed of piston of 600 feet per minute," the excess of the back-pressure above the atmosphere, in well-clothed cylinders, was 0.163 of the excess of the pressure of the steam, at the instant of exhaust, above the atmosphere.

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* Railway Machinery.

In designing, the order of study of the factors of the whole problem should be such as to settle:

(1) The engine-power, weight, and speed;

(2) The proportion of weight to be placed on the driving-wheels to secure adhesion;

(3) The size and proportions of the boiler supplying the prescribed boiler-power;

(4) The minimum safe size, and the proportions, of the steam-cylinders;

(5) The form of valve and its gear, and its desirable proportions;

(6) The details of construction.

The engine-power is fixed by the nature of the traffic. On small trains and on local roads, the maximum size and speed of regular trains determine it. On large "trunk" lines and with heavy transportation, the question is, rather: What size and cost is allowable? and this limit may even be fixed by the weight of the rails in use. Maximum hauling-power on freight trains, and maximum safe speed with passenger trains, are the objects in view.

The weight to be placed on the driving-wheels is determined by the service. Heavy, slow trains may require the whole, and eight- and ten-wheeled engines illustrate this case. Fast passenger trains on trunk lines offer a comparatively low resistance, but demand high velocities, and one, or at most two, pairs of driving-wheels meet this requirement. Both classes of engine may be rated at the same horse-power; but the one turns small wheels and moves at low speed with a tremendous tractive power; the other has wheels of 6 or 7 feet diameter, and overcomes the smaller resistance at speeds varying up to 40, or even 60, miles per hour, the weight on its drivers being two thirds the total weight of engine, or less. In any given case, the weight, W, *on the driving-wheels* must exceed the maximum anticipated pull, R, in the proportion

$$W = \frac{R}{f}; \quad R = fW;$$

in which f is the average coefficient for a wet rail, about

$$f = 0.20.$$

The size and proportions of the boiler must be such as to produce the steam needed to develop the power demanded to overcome the expected resistance at the intended speed. It must be sufficient to produce an indicated power exceeding the net hauling power, or dynamometric power in the proportion

$$I.H.P. = D.H.P. \div E;$$

when E is the efficiency of mechanism, which may be safely taken in such cases as not less than 0.80, and

$$I.H.P. = 1.25\ D.H.P.$$

It must produce from 20 to 40 pounds of steam per hour and per horse-power, accordingly as it supplies a good compound engine or a wasteful simple engine. Good engines of the latter type may be assumed to demand 25 pounds.

The steam-cylinders must be proportioned to exert the full maximum tractive power demanded at starting, and to work off the steam made by the boiler with a ratio of expansion of about 2.

The weight of fuel burned in the furnace of the locomotive often exceeds 60 pounds on the square foot of grate, and is sometimes forced up to double this quantity. Exceeding about 75 pounds, however, the waste of unburned fuel thrown out of the furnace becomes excessive. A loss of 10 per cent by this action is common, and much more often takes place. The quantity of water evaporated is usually not much above 6 times the weight of fuel, although 7 and 8 are unquestionably, and possibly even 9 sometimes, attained; but reports of higher figures are usually based upon tests in which the quality of the steam was not determined; and the presumption is, in such cases, that the apparent gain is really due to "priming."

The volume of water contained in the boiler at its usual gauge may be evaporated in from 20 to 45 minutes.

The power developed may be reckoned as 30 I. H. P. per square foot of grate, as a maximum, and probably not over 20 in ordinary practice; while 15 is a safer figure to be used in designing, and may be taken, probably, as a fair minimum. From 2.5 to 5 pounds of fuel are used per I. H. P. per hour in

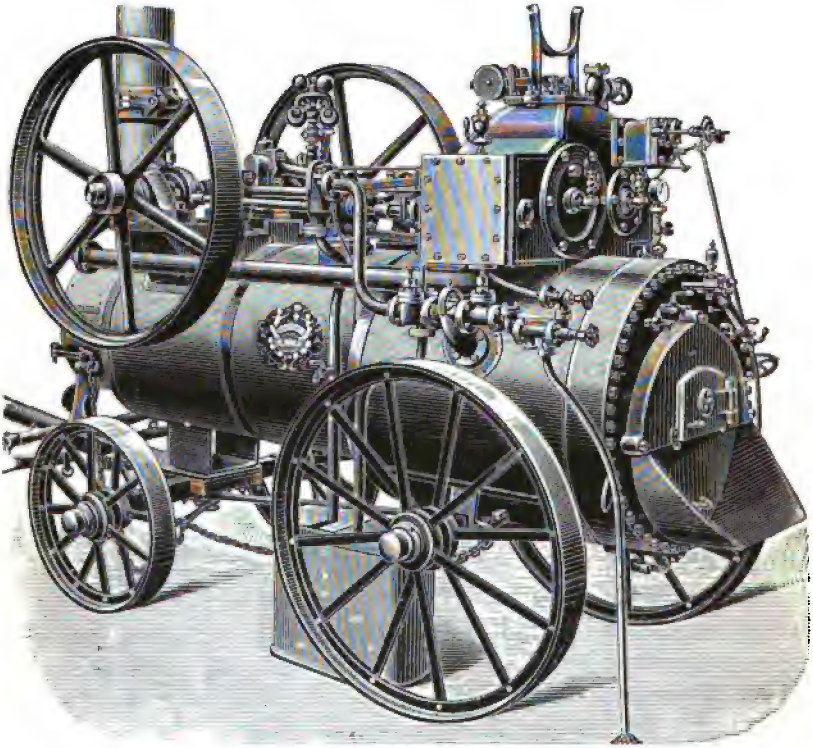

FIG. 56.—PORTABLE RECEIVER COMPOUND ENGINE.

good practice, the minimum figure being obtained with later compound engines.

The valves and their driving mechanism must be made to give the required distribution of steam at minimum cost of construction and maintenance, and with the least practicable expenditure of power.

All details of construction must be simple, strong, and durable, and as light, out of the way, and safe against accident

as is possible. There must usually be a leading pair or system of "pilot" or guiding wheels. The wheels, whether in two or more groups, or constituting a single group, must carry the boiler and "running-gear" by means of a system of equalizing levers which will distribute both the regular loads and the incidental disturbances of the engine moving on the track. Thus this system of suspension becomes an essential characteristic of this type of the steam-engine.

The general design of the portable engine is exhibited in the next figure, as adopted by Wolf of Magdeburg. It includes the removable tubes elsewhere described, and a very compact arrangement of machinery, the boiler serving as its foundation and, in part, its frame. The engine is compound, and the comparison of this design with those of the British and American builders will exhibit the essential differences of similar designs as built in the several countries.

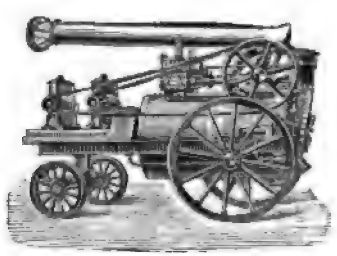

FIG. 57.—"DYNAMO" ENGINE.

The little sketch here given exhibits the Wolf system of mounting engine and "dynamo" in portable form for transportation. The design has been found very satisfactory in many cases, and illustrates the readiness with which the engineer may adapt his design to special purposes.

30. **Stationary-engine** designing involves, as a rule, fewer incidental problems than the planning of any other type. The conditions prescribed are usually simpler and are more definite than in marine- or locomotive-engine designing. The choice of steam-pressure, that of engine-speed, of type of engine, or of methods of connection with the machinery to be driven, are all given more latitude than in any other case. General practice and experience has, however, settled down to the use of engines either of long stroke and moderate speed of revolution, with small clearance, separated steam- and exhaust-valves, and a detachable valve-motion momentarily and automatically adjusted by the governor, or of the now common "high-speed"

variety, with short stroke, great velocity of rotation, and an exceedingly delicate and effective form of "shaft-governor." The former is the standard mill-engine, the latter that adopted especially for fast machinery.

It often happens, however, that only a very careful comparison of the results of engine-trials and of estimates of costs will decide which is, for a specified place and purpose, the better. The method of collating and of employing the data are elsewhere given in detail.

The weights of stationary engines of usual types are variable, but the following are given by Haswell as usual:

WEIGHTS OF STATIONARY ENGINES.

Vertical beam, geared to mill,	9600	lbs. per cu. ft. cyl.
" " " "	4290	" " "
Horizontal..................	5100	" " "
"	5600	" " "

The sizes varied from 14 to 30 inches diameter of cylinder, and from 2 to 5 feet stroke of piston.

The general and complete design of a stationary engine of recent construction, and intended for use in electric-light operation and at high speeds of rotation, is well shown in the accompanying sectional drawing of an engine constructed by Mr. Ide. The direct connection, strength of running-parts, and enclosing crank-case are elements of importance where, as in such engines, safety, solidity, and certainty of lubrication are essential to successful employment. This method of securing flooded journals has been already seen in the sketches of the Westinghouse engines.

Small engines are often preferred vertical, as in Fig. 59; since they occupy less floor-space, are quite as accessible as if horizontal, and are readily placed in any convenient locality. In many cases it has now become customary to use in these engines the shaft-governor, as here shown, controlling the speed by its action on a single valve, commonly of the piston variety, and always, necessarily, some form of balanced valve.

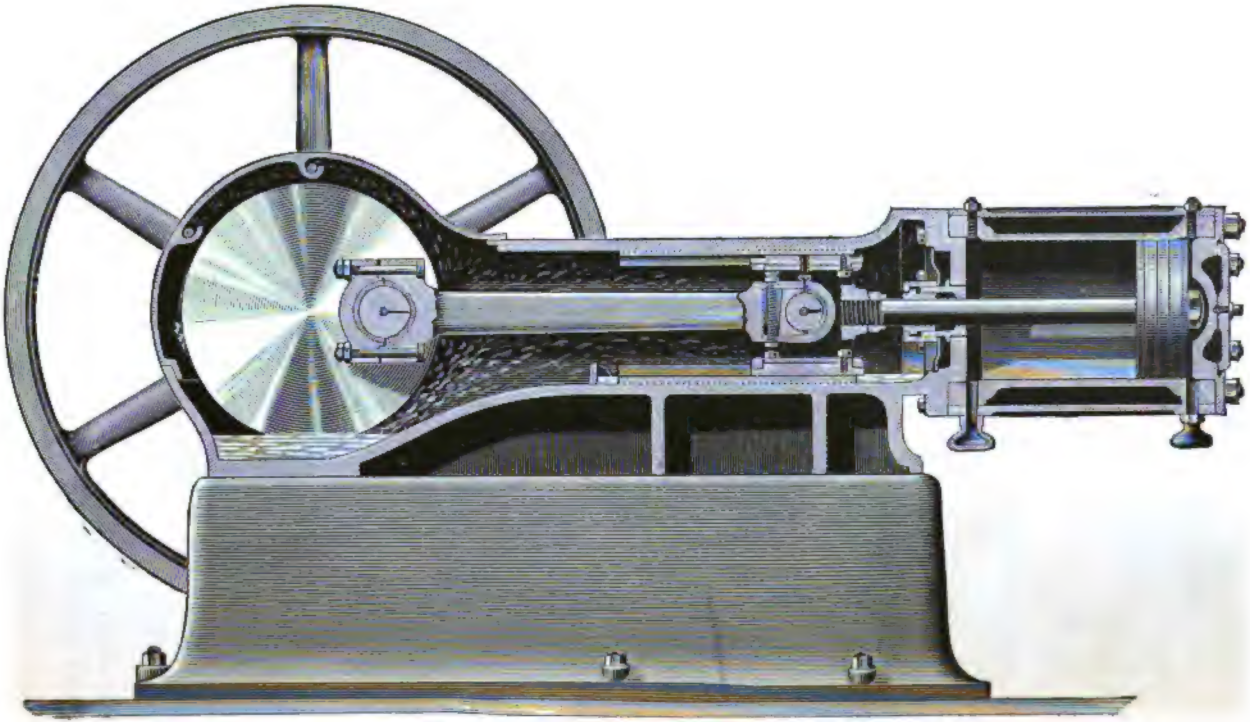

FIG. 58.—SECTION OF STATIONARY ENGINE.

The design of the Dow steam-turbine, to which reference has been made (Vol. I. § 44, p. 237 *et seq.*), is illustrated in

FIG. 59.—VERTICAL ENGINE WITH SHAFT-GOVERNOR.

Figs. 60 and 61; the former showing the axial, the latter the transverse, section.*

In the former, *aa* is the casing, and the bonnets or covers, *bb*, are so constructed and secured as to permit accurate adjustment of the disks, *cc*, and the body of the cylinder. On the

* From "Power," May 1891.

faces of these disks are the guide-blades, spirally arranged, as better seen in the second sketch. Steam enters through the passage *ee*, and finds exit through *fff* to the exhaust-pipe, *g*.

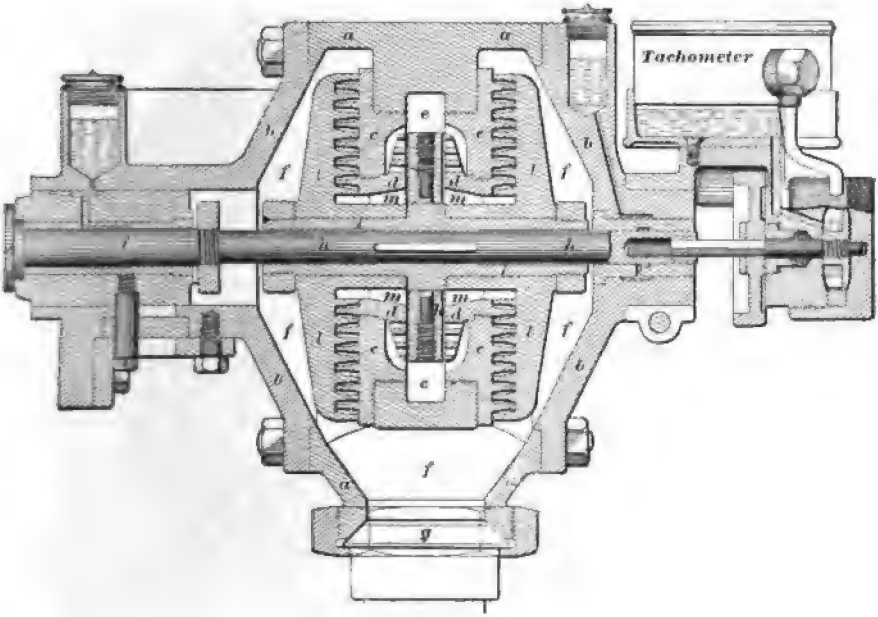

FIG. 60.—DOW STEAM-TURBINE.

The main shaft, *hh*, is carried on journals at each side the casing, as seen, and a sleeve, *ii*, stiffens the central part of the shaft and carries the turbine-wheels, proper, *ll*, of which a pair are used to insure a longitudinal balance of pressures. These are "~~inward~~ Outward-flow" turbines, "compounded" by having a number of concentric circles of blades working in series, in conjunction with the guide-blades, on *cc*, as is well shown in the last figure.

Steam entering from *ee* must pass through the balance-disk *k*, on the shaft, the spaces on either side, and the passages *mm*, to reach the turbine-disk. This keeps the sleeve and the three disks automatically adjusted, longitudinally, at the intended very minute distance from the guide-disks, and insures that contact shall not take place. The sleeve is splined to the shaft,

and permits slight endwise motion of the latter without affecting the action of the turbine.

FIG. 61.—DOW TURBINE.

31. Pumping-engine designing involves, in addition to the engine proper, the proportioning of a set of pumps to their work, their connections with the supply and delivery mains, and their attachments to the engine. The forms of the standard engines of this class have been fully illustrated. They must usually be made economical in their use of fuel, and their pumps must be given the highest possible modulus of efficiency. It is therefore customary to adopt high steam-pressure, a maximum ratio of expansion, and jacketed cylinders, to take special care to insure the supply of dry or superheated steam, and to give their main pumps moderate speed and very large valve area. In all other respects the methods of design are those adopted for all types of engine of high efficiency.

The more steadily the water can be moved, the less the power required and the cost of its delivery. A double-acting pump and large air-vessels are therefore commonly employed, and these pumps, even, are often worked in sets of two or more. By the use of two such pumps, the variation of velocity from the mean is reduced from ten per cent to about five, and three pumps in "multiple arc" reduce that variation to 2.5 per cent, nearly. An air-chamber of at least ten times the capacity of a single pump should be employed. It should be still larger under high pressures, and should always be provided with convenient means of supplying it with air.

The whole design is necessarily subordinated to the demands of the pump part; and the weight, inertia, and friction resistances of the water are the first and most essential matters to be studied. The rate of flow in pumps, passages, valve-chambers, and mains should not exceed five feet per second, and would be better four; while the velocity of pump piston is rarely made much in excess of 200 feet per minute, or 3.33 feet per second. Reflux of water should be carefully guarded against at every step, and especial care taken that a long column of water in mains once in motion, in either direction, shall not be suddenly checked, as it is certain to result in the bursting of the mains. Ample safety-valves, or standpipes, should always be provided where such a contingency is possible.

Pumps should be constructed, as directed in the article in which their design was considered, with large valve-area, and preferably with numerous small valves rather than a few of large size. Care should be taken to provide facilities for convenient renewal of all packing and replacement of all valves.

The "slip" of the pumps of pumping-engines usually falls between 2 and 3 per cent. The Corliss engine at Providence, R. I., is said to have come down to 0.5 per cent; but a more usual figure, on familiar types of engines, is 8 or even 10 per cent.

In this design, as here seen, the whole system is made light, free from friction, with carefully designed cylinders and pumps, and with every care to reduce the several now well-recognized

wastes to a minimum. It is an excellent illustration of a good general design.

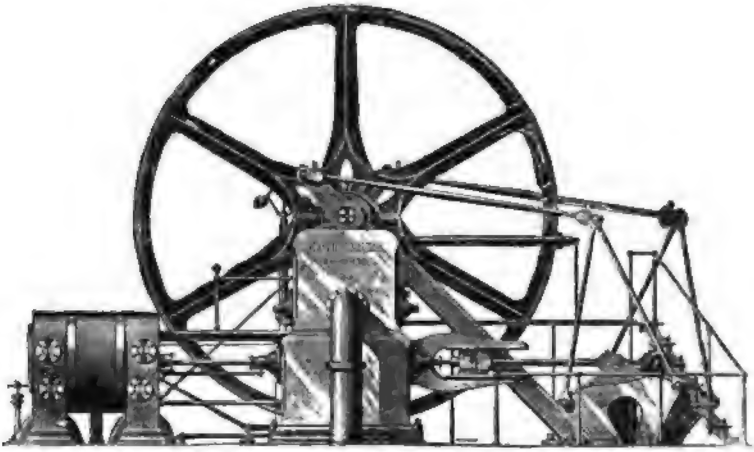

FIG. 62.—CORLISS'S PAWTUCKET ENGINE.

32. **The Design of the Steam-boiler** involves a careful preliminary study of the strength and other physical characteristics of the materials available, of the fuels to be employed in the production of steam, and of all those conditions which dictate a choice of type and the location of the steam "plant." These conditions have been already discussed in brief, and will be found more fully developed in special treatises on the steam-boiler, which should be consulted when taking up this branch of engineering. *

The location of a boiler is sometimes a matter of choice with the engineer preparing the plans, and may be one of serious importance. Where possible it should always be so chosen that the boiler may be easy of access for inspection and repair; it should be free from special danger to lives or surrounding property in case of accident; and the site selected should be dry and well protected against the weather. The nearer the engine or other point at which its steam is delivered, the better. Only sectional boilers should be placed under

* For a more detailed treatment, see A Manual of Steam-boilers, by the Author; N. Y., J. Wiley & Sons; from which this chapter is abstracted.

buildings. Shell-boilers should have boiler-houses constructed for them apart from the larger and more important structures to which they are auxiliary, and this precaution is especially advisable for cases—as mills—in which many lives may be endangered. The risk involved is not great where these boilers are well designed and constructed; but the prudent engineer avoids even moderate risk where a life is involved.

When the space is restricted in floor-area, but of good height, the upright tubular boiler is selected; if the floor-area is unrestricted, but head-room is small, the horizontal forms of boiler are chosen. Good forms of "safety" boilers may be placed wherever they can be given room, provided they are accessible for inspection, cleaning, and repairs.

The choice of fuel and of method of combustion is commonly necessarily made before the design can be proceeded with. The fuel is, as a rule, selected mainly with a view to commercial efficiency; but the presence of any observable quantity of sulphur in coal justifies its rejection at even considerable pecuniary sacrifice. That fuel is best which produces the required quantity of steam with certainty and regularity under the given conditions, and at minimum total cost for purchase, transportation, and handling, storage, interest, and insurance, and wear and tear of apparatus.

The combustion may be produced by either a natural chimney draught, or a forced draught created by a fan, a steam-jet, or other artificial means. With very fine coal, or where the grate-area or the boiler itself is so small as to make the rate of combustion due to natural draught insufficient, the forced blast is employed. Natural draught is to be preferred where the desired amount of steam may be made by that system.

The conditions of efficiency in steam-boilers are those affecting the production, the transfer, and the storage of the heat-energy derivable from the fuel. These have been considered elsewhere. *En résumé:* the efficient production of heat requires the concentrated combustion of the fuel, with the minimum air-supply consistent with the complete combination of its oxidizable elements with oxygen, and the attainment of

maximum temperature. The efficient transfer and storage in the steam of this heat demands that it be liberated at maximum temperature, that the heating-surfaces be of great extent in proportion to the weight of fuel burned and to the quantity of heat liberated, and that these surfaces be effective in absorption of heat.

33. **The Principles of Design,** in the case of the steam-boiler, involve those of strength of materials and of structures, the determination of the size, form, and proportions of parts; the relation of area of heating and of grate surface to fuel burned; the character and proportions of accessory parts; in fact, the application of all the data and the laws which have been studied in the preceding portions of this work. The designing engineer must determine the form and proportions of a vessel in which is to be generated a given quantity of steam with satisfactory efficiency and safety, and with as nearly permanent commercial success as possible.

The settlement of the general proportions of the structure is made with reference to the above considerations; but general experience has brought these proportions into a fairly definite relation, and, as an illustration, the better classes of boiler rarely have a less ratio of heating to grate surface, where natural draught is adopted, than about 25 to 1, or a higher ratio than 40 to 1. With more intense combustion and forced draught this proportion is considerably increased. The best proportion is probably usually capable of fairly exact calculation by a method to be considered at some length in a later chapter. Boiler-power is very often calculated, in cases of ordinary practice, by allowing a certain number of square feet of heating-surface to the horse-power. Thus, the following may be taken as a fair average set of figures:

Plain cylinder-boiler	8
Flue-boiler	10
Water-tube or sectional boiler	12
Locomotive boiler	13
Return tubular boiler	15
Upright tubular boiler	18

Careful calculation should be resorted to in every important case.

In designing boilers the effort of the engineer should be—

(1) To secure complete combustion of the fuel without permitting dilution of the products of combustion by excess of air. A combustion-chamber is usually desirable.

(2) To secure as high temperature of furnace as possible.

(3) To so arrange heating-surfaces that, without checking draught, the available heat shall be most completely taken up and utilized and the most complete and rapid circulation secured, both for the water and for the furnace-gases.

(4) To make the form of boiler so simple that it may be constructed without mechanical difficulty or excessive expense, and to arrange for ample water-surface, as well as large steam and water capacity, so as to insure against serious fluctuation of steam-supply.

(5) To give it such form that it shall be durable, under the action of hot gases, and of corroding elements of the atmosphere.

(6) To make every part accessible for cleaning and repairs.

(7) To make all parts as nearly as possible uniform in strength, and in liability to loss of strength with age, so that the boiler, when old, shall not be rendered useless or dangerous by local defects.

(8) To adopt a reasonably high "factor of safety" in proportioning parts, and to provide against irregular strains of all kinds.

(9) To provide efficient safety-valves, steam-gauges, mud-drums, and other appurtenances.

(10) To secure intelligent and very careful management.

In securing complete combustion—the first of these desiderata—an ample supply of air and its thorough intermixture with the combustible elements of the fuel is essential; for the second—high temperature of furnace—it is necessary that the air-supply shall not be in excess of that absolutely needed to give complete combustion. The higher the furnace-temperature,

and the lower that of chimney, the greater the proportion of available heat.

It is further evident that, however perfect the combustion, no heat can be utilized if either the temperature of chimney approximates to that of the furnace, or if the temperature of the furnace is reduced by dilution to that of the chimney.

In arranging heating surface, the effort should be to impede the draught as little as possible, and so to place them that the circulation of water within the boiler should be free and rapid at every part reached by the hot gases.

The direction of circulation of water on the one side and of gas on the other side the sheet should, whenever possible, be opposite. The cold water should enter where the cooled gases leave, and the steam should be taken off farthest from that point. The temperature of chimney-gases has thus been reduced by actual experiment to less than 300° Fahr., and an efficiency equal to 0.75 to 0.80 the theoretical is attainable.

The extent of heating-surface simply, in all of the best forms of boiler, determines the efficiency, and the disposition of that surface seldom affects it to any great extent. The area of heating-surface may also be varied within very wide limits without greatly modifying efficiency.

The factor of safety is usually too low. The boiler should be built strong enough to bear a pressure at least six times the proposed working-pressure. As it grows weak with age, it should be occasionally tested to a pressure at least double the working-pressure, which latter should be reduced gradually to keep within the bounds of safety.

34. **The Controlling Ideas** *in designing* dictate the following procedure. The engineer determines—

(1) The height of chimney, and rate of combustion desirable or practicable.

(2) The type of boiler, having regard to the character of water to be used as "feed," and the costs of construction, operation, and maintenance.

(3) The quantity of steam that will be demanded.

(4) The efficiency of boiler that it will be economical to se-

cure, according to the principles to be given, and thus the ratio of heating to grate surfaces.

(5) The kind and the quality of fuel required, with the given or proposed efficiency, to produce the demanded quantity of steam.

(6) The total areas of grate and of heating surface required to burn that fuel and to make that steam.

(7) The forms, sizes, and proportions of details.

The dimensions and proportions of the boiler plant being thus determined, the engineer decides what amount of power shall be obtained from a single boiler, and thus how many boilers are to be constructed, the area of heating and grate surface to be given each; and he finally decides upon the form of setting, and method of making steam and water connections.

It then remains only to make a drawing of the boiler, which shall show its form and dimensions, the arrangement of stays, pipes, safety and other attachments, and the setting. The first plan constructed will usually require some modification to adapt it exactly and satisfactorily to the wants of the user; which changes being made, the boiler may be constructed from the drawing. The thickness of shell, size of tubes or flues, sizes, methods, and distribution of stays, and similar matters of detail, are settled by well-known rules of practice, or by the consideration of the peculiar conditions met with in the case in hand.

The Steam-pressure to be adopted will necessarily be one of the first matters to be considered and settled; both because it has an important bearing upon the efficiency of the engine, and because it must be kept in view in the selection of the type and size of boiler. The tendency is constantly in the direction of higher steam-pressure, and the consequent adoption of the simpler, stronger, and safer kinds of boiler. This directly conflicts with the commercial considerations affecting boiler construction, especially of the common forms of shell-boiler. The larger the boiler, as a rule, the cheaper, comparatively, its construction, the less the cost of setting and of installation, and the higher its economy in operation. A large shell, however,

must be made of thicker iron, and is always somewhat less absolutely safe than a similar smaller structure.

A limit is thus being continually approached because of the fact that the net gain is less and less as the increase occurs at higher pressures.

An ingenious system of boiler-construction, devised by Wolf of Magdeburg-Buckau, is that here shown, in which the whole "nest" of tubes is removable for inspection or for "scaling."

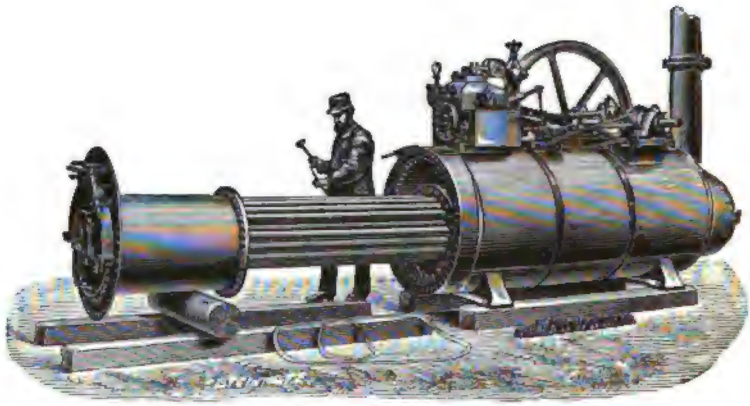

FIG. 63.—REMOVABLE TUBES.

The joints between heads and shell are made with bolts and suitable packing; and an advantage of real importance, in many cases, is thus obtained.

35. **The Size and Design of the Chimney,** its height and area of flue, are modified somewhat by its form and proportions, and by the character of its interior surfaces. The greater the friction-head, the less its effectiveness. A chimney of circular section and with a straight uniform flue is better than with any other section or with less direct flue. The flue-area is either uniform or tapering toward the top, in which latter case the area for calculations is measured at the top. Mr. Kent assumes that the friction may be taken as equivalent to a reduction of section of two inches all around, and a square flue section as equivalent to a circular one of diameter equal to

its side.* He thus obtains the following: Assuming a commercial horse-power to demand the consumption of 5 pounds of coal per hour, we have the following formulæ:

$$E = \frac{0.3\,HP}{\sqrt{H}} = A - 0.6\sqrt{A}; \quad . \quad . \quad . \quad . \quad (1)$$

$$HP = 3.33E\sqrt{H}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$s = 12\sqrt{E} + 4; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$d = 13.54\sqrt{E} + 4; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$H = \left(\frac{0.3\,HP}{E}\right); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

in which HP is the horse-power; H is the height of chimney in feet; E the effective and A the actual section, in square feet; s is the side of a square, and d the diameter of a round, flue, in inches.

36. Forms and Proportions of Furnace and Grate are settled upon so soon as the character of the fuel and the proportions of chimney are fixed.

The rate of combustion is fixed as a maximum, as already seen, by the height of chimney.

Fuels should evaporate, respectively, from feed-water at the boiling-point and at atmospheric pressure, under the most favorable possible conditions, about as follows:

	Relatively.	Weight water per Unit Weight of Fuel.
Best anthracite..........................	100	13.5
Best semi-anthracite and bituminous..	110	15
Ordinary coals, soft..................	80	11
Ordinary coals, anthracite............	75	10

* Trans. Am. Soc. M. E., 1884.

Examples of these several classes are seen in the best Pennsylvania anthracites, the Welsh and Maryland semi-anthracites, or semi-bituminous coals, the ordinary good bituminous fuels of Nova Scotia and of Western Pennsylvania, and the earthy coals of the West.

The quantity of steam actually made will depend upon the temperature of the feed-water, and will be less as the water is colder. It is customary, as elsewhere stated, to reduce the results of experiments determining efficiency of boilers to "equivalent evaporation from and at the boiling-point," under atmospheric pressure.

When the maximum possible evaporation is given for feed at 212° F. (100° C.), and at atmospheric pressure, i.e., under the standard conditions, multiplying that figure by the reciprocal of the factor of evaporation for the proposed temperatures of feed and of steam will give the maximum possible evaporation under the latter conditions. Thus we get the following:

RELATIVE EVAPORATION AT VARYING TEMPERATURES OF FEED.

Temperature of feed-water...	212° F.	200	180	160	140	120	100	80	60	40
	100° C.	93.3	82.2	71.1	60.0	48.8	37.7	26.6	15.5	4.4
Relative steam evaporation..	100	98	96	94	92	90	88	87	86	84

The coals in common use in the United States are:

The semi-bituminous coals from Maryland;

The anthracites from Pennsylvania;

The bituminous coals from Pittsburg and Western Pennsylvania;

The bituminous coals from Ohio and the West.

When burned in ordinary furnaces, these coals will make steam, per pound of coal, in nearly the following proportions, as given by Mr. T. Skeel:*

Semi-bituminous	110
Anthracite	100
Pittsburg	90
Ohio	75

* Weisbach, vol. II.

The weights that may be burned on the same grate, with the same chimney, will vary nearly as follows:

Anthracite	100
Semi-bituminous	120
Pittsburg	120
Ohio	200

Relative areas of grate-surface that will be necessary to burn coal enough to furnish the same quantity of steam are nearly as follows:

Anthracite	100
Pittsburg	90
Semi-bituminous	75
Ohio	67

This refers to the average coal of each kind in practice.

The loss as refuse falling through well-proportioned grate-bars may be taken as 5 to 10 per cent for good bituminous coals, or 10 to 20 per cent for the lower grades, and about the same for anthracites. Wood may be taken by weight as having one half the value of coal. A cord of best hard wood should equal a ton of good coal.

From the results of chemical analyses, the evaporative power of various kinds of fuel, expressed in pounds of water per pound of fuel evaporated from and at 212° F., which we will call E, has average values given by Prof. C. A. Smith * in the following table, which may be found useful as supplementary to the several other sets of data already given in this connection:

Kinds of Fuel.	E
Pure carbon completely burned to CO_2	15
Pure carbon incompletely burned to CO	4.5
CO completely burned to CO_2	10.5
Charcoal from wood, dry	14
Charcoal from peat, dry	12

* Am. Engineer, 1883.

Kinds of Fuel.	*E*
Coke good, dry	14
Coke average, dry	13.2
Coke poor, dry	12.3
Coal, anthracite	15.3
Coal, dry, bituminous, best	15.9
Coal, bituminous	14
Coal, caking, bituminous, best	16
Coal, Illinois (from four mines near St. Louis)	12
Lignite	12.1
Peat, dry	10
Peat with one fourth water	7.5
Wood, dry	7.25
Wood with one fifth water	5.8
Wood, best dry pitch-pine	10
Mineral oils, about	22.6

The computation of the heat of combustion of a gas of which the chemical composition is given is simply as follows, the multipliers being each the heating-power of 0.01 cubic foot respectively. Assume an analysis thus:*

	Comp.		Heat-coefficient.		B. T. U. per cu. ft.
Nitrogen	2.10				
Carbon dioxide	1.00				
Illuminating gases	5.19	×	16.82	=	87.3
Carbon monoxide	7.50	×	3.43	=	25.7
Marsh gas	34.00	×	10.73	=	364.8
Hydrogen	50.21	×	3.44	=	172.7
					650.5

Or, about 650 British thermal units per cubic foot completely burned.

37. **The Relative Areas of Chimney, Flues, and Grate** are seen to be variable with the circumstances under which the boiler is to be operated, but with natural draught and usual

* American Gas-light Journal, March 9, 1891.

working conditions certain proportions have become almost universally accepted as standard in common practice. Thus it may be taken as well settled by experience, that in chimneys of circular section, smooth internal surfaces, and in the open, where draught is unobstructed by air-currents produced by surrounding objects, as, for example, with marine steam-boilers, the minimum ratio of chimney-flue section, section through the tubes and that over the bridge-wall to grate-surface, should be, at least, respectively, $\frac{1}{9}$, $\frac{1}{8}$, $\frac{1}{7}$, while a maximum to be adopted with forced draught is not far from $\frac{1}{7}$, $\frac{1}{6}$, and $\frac{1}{5}$, for anthracite coal. The latter ratios will also work well for bituminous, free-burning coals and natural draught; and the sections may often be made still greater with advantage, when a blast is also used with such fuel.

With restricted draught-area the amount of fuel that may be burned becomes reduced. Thus, assuming a chimney 50 feet (16 m.) high:

Area of least flue-section (grate = 1)....	0.14	0.10	0.07	0.05	0.04
Relative coal burned..................	1.	0.8	0.7	0.6	0.4
Average fuel, lbs. per sq. ft. grate......	15	12	10	9	6
" " kilogs. per sq. m.........	7.5	6	5	4.5	3

For square sections of chimney-flue and with rough interior surfaces the size of chimney is increased both in weight and area of section. As a general rule, the height of factory chimneys is increased with the size and number of boilers, irrespective of the above-stated ratios, and a not uncommon proportion of "stack" is that which makes the height about twenty times the diameter of the flue. Ordinary mill-chimneys, for moderate powers, range between 50 and 100 feet (16 and 32 m.) in height.

38. Common Proportions of Boiler are found in ordinary practice to be not far from those given below.

The interior space of the boiler is commonly divided into about two thirds or three fourths water-space, the remainder being steam-room. In marine boilers more steam-space should be given.

RATIO OF HEATING- TO GRATE-SURFACE.

Plain cylinder boilers	12 to 15
Cornish	15 to 30
Cylindrical flue	20 to 25
" tubular	25 to 35
Marine tubular (fire)	30 to 35
" " (water)	35 to 40
Locomotive tubular	50 to 100

The ratio of heating- to grate-surfaces should, where possible, be always carefully determined with reference to maximum commercial efficiency in the manner described in a later chapter.

The above proportions produce ratios of weights of fuel burned per unit area of heating-surface, in general practice, about as follows:

RATIO OF FUEL BURNED TO HEATING-SURFACE.

	Pounds per sq. ft. H. S.	Kilogs. per ~~sq. m.~~
Stationary boilers	0.5 to 1.0	0.1 to 0.2
Marine (natural draught)	0.5 to 0.6	0.1 to 0.3
Locomotive and forced draught	0.8 to 1.0	0.4 to 0.5

Similarly, the power of such boilers may be reckoned roughly as below, and their relative standing in efficiency and capacity taken as follows:

HORSE-POWER AND ECONOMY.

	Per H. P.		Relative Standing.	
	Sq. ft.	Sq. in.	Capacity.	Economy.
Water-tube	10 to 12	1.0 to 1.1	1.	1.
Fire-tube	14 to 18	1.3 to 1.6	0.75	0.9
Flue	8 to 12	0.7 to 1.1	0.50	0.8
Plain cylindrical	6 to 10	0.5 to 0 9	0.20	0.7
Locomotive	1 to 2	0.1 to 0.2	0.6	0.8

The above, as with every proportion and detail of the steam-boiler, should always be made the subject of careful calculation whenever the case is in the least degree peculiar.

The following are proportions frequently accepted by the trade for one of the most common varieties of stationary boiler sold in the market:

PROPORTIONS OF CYLINDRICAL TUBULAR BOILERS.

Number of Size.	Horse-power.	Diameter—inches.	Length—feet and inches.		Flues.			Dome.		Thickness of Shell—inches.	Thickness of Head—inches.	Length of Furnace—feet.	Stack.		Weight of Boiler—pounds—about.	Weight of Boiler and Fixtures—pounds. Complete.
					Number.	Diameter—inches.	Length—feet.	Diameter—inches.	Height—inches.				Diameter—inches.	Height—feet.		
1	15	36	8	11	30	3	8	20	20	¼	[illegible]	3	18	26	2,950	5,350
2	20	36	10	11	30	3	10	20	20	¼	[illegible]	3½	18	30	3,500	5,900
3	25	42	11		38	3	10	24	24	9/32	[illegible]	3½	20	30	4,400	7,100
4	30	42	13		38	3	12	24	24	9/32	[illegible]	4	20	36	5,000	7,800
5	35	44	13		46	3	12	24	26	5/16	[illegible]	4	22	36	5,500	8,700
6	40	48	13	2	52	3	12	24	28	5/16	[illegible]	4	24	36	6,400	9,900
7	45	50	14	2	52	3	13	30	30	5/16	[illegible]	4	24	36	6,800	10,400
8	50	54	13	2	58	3	12	30	30	5/16	[illegible]	4	26	36	7,600	11,500
9	60	54	16	2	58	3	15	30	30	5/16	[illegible]	4½	26	45	8,550	12,750
10	70	60	15	4	76	3	14	30	30	11/32	7/16	4½	28	45	10,000	14,500
11	75	60	16	4	76	3	15	30	30	11/32	7/16	4½	28	50	10,500	15,100
12	80	60	17	4	76	3	16	30	36	11/32	7/16	5	28	55	11,200	16,100
13	90	66	16	5	100	3	15	36	36	[illegible]	7/16	5	32	55	13,500	19,100
14	100	66	17	5	100	3	16	36	36	[illegible]	7/16	5	32	55	14,200	19,800
15	125	72	17	6	132	3	16	36	36	7/16	[illegible]	5	36	60	17,200	24,000

The upright tubular boiler is given less heating-surface than the above, is much lighter, and is less economical. The locomotive type of stationary boiler has about the same weight as the above, but rather less heating-surface.

39. **The Usual Rates of Evaporation** and the effect of varying the proportions of tubes has been well determined by the experiments of Isherwood and others.

The proportions of flues and tubes vary somewhat in practice; but it will be found seldom advisable to make tubes more than 50 or 60 diameters in length. Where the heating-surface consists principally of tubes, the efficiency will be found to vary with their length nearly as follows:

Length of tube (diameters)	60	50	40	30	20
Water per unit weight of fuel	12	11	10	9	8

When the ratio of heating to grate-area was 25 to 1, Isherwood found the evaporation to vary thus:

Fuel per hour........	8	10	12	16	20	24
Evaporation..........	10.5	10.1	9.5	8.2	7.3	6.8

which series is represented by

$$W = \frac{21}{\sqrt{F}}, \text{ nearly.}$$

Clark obtained with locomotives an equal evaporation with

Fuel (coke).............	15	25	38	56	76	98	125	153
Ratio of H. S. to G. S....	30	40	50	60	70	80	90	100

the evaporation being constant at 9 of water to 1 of fuel, which may be expressed by

$$S = 8\sqrt{F}, \text{ nearly,}$$

S being the ratio of the two areas and F the weight of coke burned on the unit of area of grate.

In estimating area of heating-surfaces the whole surface exposed to the hot furnace gases is reckoned. The formula for efficiency already given illustrates the progressive variation of the evaporative power with change of proportions of boiler.

40. **The Relation of Size of Boiler to Quality of Steam** demanded is one that occasionally becomes worthy of consideration. Where the steam is required for driving steam-engines, it is very important that it should be thoroughly dry, and it is an advantage to moderately superheat it. Maximum economy cannot be attained where wet steam is used. A boiler attached to a steam-engine, and especially where fuel is costly and efficiency important, should have ample heating-surface, some superheating-surface if practicable, ample extent of water-surface area to permit free separation of steam and water, and large steam-space.

Steam employed for heating purposes is not necessarily dry; it may carry a large amount of water with it into the system of heating-coils or radiators, and yet give good results, if the latter are of large section. Where the pipes are of restricted area of section, however, wet steam flowing less freely than when dry or superheated, there may result such a retardation of flow and of circulation as may cause considerable increase of cost. This has been found sufficiently great, in some cases, to justify drying, and perhaps superheating, the exhaust-steam from engines where used for heating purposes. As a general rule, the boiler must be made a trifle larger to supply perfectly dry steam and do good work.

In the use of steam for heating purposes, one square foot of boiler-surface will supply from 7 to 10 square feet of radiating-surface. Small boilers should be larger proportionately than large boilers. Each horse-power of boiler will supply from 250 to 350 feet of 1-in. steam-pipe, or 80 to 120 square feet of radiating-surface.

Under ordinary conditions one horse-power will heat about—

Brick dwellings, in blocks, as in cities..	15,000	to	20,000	cu.	ft.
" stores " "	10,000	"	15,000	"	"
" dwellings, exposed all around....	10,000	"	15,000	"	"
" mills, shops, factories, etc........	7,000	"	10,000	"	"
Wooden dwellings, exposed	7,000	"	10,000	"	"
Foundries and wooden shops..........	6,000	"	10,000	"	"
Exhibition buildings, largely glass, etc..	4,000	"	10,000	"	"

41. **The Number and Size of Boilers** to be used in any case in which considerable power is demanded is determined mainly by practical considerations related to their construction. As a rule, the larger boiler is more economical in first cost and in operation, within certain limits, than several smaller boilers of equal aggregate power. But passing a limit which cannot be usually very exactly defined, expense is increased, transportation becomes difficult, location and setting involve problems difficult of solution, and management becomes less easy.

Mr. Leavitt has, however, constructed stationary boilers, of a peculiar modification of the locomotive type, of as high as one thousand horse-power; and marine boilers of equal or greater power have been built not infrequently for steamers plying on the larger rivers of the United States. Stationary boilers of 100 horse-power and marine boilers of 500 are more usual and more commonly suitable sizes. Locomotive boilers are necessarily always sufficiently large to supply all the power demanded of the engine.

The type of boiler has much influence on the limit of size. Plain "cylinder boilers" are rarely made more than from 3 to 4 feet (0.9 to 1.2 m.) in diameter, and this restricts the grate-area so that the power derivable from a single such boiler is seldom more than 15 or 20 horse-power, and is usually much less. The more complex structures often include several furnaces, and yield from 100 to 200 horse-power each on land, and more at sea.

Makers in the United States usually allow 15 square feet of heating-surface and one of grate to the horse-power, in plain cylindrical boilers, and the same area of heating-surface, but a fourth and a half less grate-area, respectively, with flue-boilers and tubular boilers, where estimating for the market.

M. de Pambour found the priming of French locomotive boilers in 1834 to amount to about 30 per cent; M. de Chatellier, in 1843–4, found it to be 30 to 50 per cent: but a large proportion of the moisture measured was undoubtedly the product of cylinder-condensation.

Builders of the more economical classes of engines supply them with boilers often of less size than the accepted standard rating would dictate, as they demand less steam per horse-power than the average engine. A good engine of moderate size, with an automatically governing and adjusting valve gear, if condensing, should give good results on as low as seven or eight square feet of heating-surface per actual horse-power, and, if non-condensing, with ten or twelve square feet. Large engines are given a smaller allowance of heating-surface, proportionally, than are small engines.

42. **The Standard Sizes of Tubes** have become well settled by custom. So large an element of boiler-construction necessarily assumes, with time, a somewhat rigid set of proportions. The sizes employed range from 1 or 1¼ inch (25.4 to 31 mm.) diameter in the smallest boilers, to 2 or 2½ inches (51 to 63.5 mm.) in the locomotive and other boilers of moderate size; and to 3 or 4 inches (76 or 102 mm.), or even 5 or 6 inches (1.27 or 1.52 mm.), in large boilers, or where a very free draught or greater convenience of access are required. Water-tube boilers are commonly given tubes 4 or 5 inches (102 or 127 mm.) in diameter. The length of the tube is customarily not above 50 or 60 diameters in stationary boilers, and two thirds this length in marine work. The spaces between the tubes should be about one half their diameter; they are, however, usually placed much closer. All tubes in our market are gauged to British measures, as below.

When the dimensions of a tubular boiler are given, the outside diameter of the tubes is usually stated, so that twice the thickness must be subtracted to obtain the diameter to be used in the calculation of heating-surface. The thickness of tubes by different makers varies somewhat, but those given below are average values, and can be used without serious error.

The dimensions of standard tubes as made by some of the best makers in the United States are given in the table on the next page.

In a flue-return tubular boiler the area of flues should be about 20 per cent, and the draught-area of uptake about 25 per cent greater than the draught-area of tubes. Good conditions for combustion and steaming are realized when the grate-surface is 8 times and the heating-surface about 200 to 240 times the draught-area of tubes.

The location and arrangement of fire-tubes has an important bearing on the distance by which they may be safely separated. In locomotive boilers, where they only check the rise of currents laden with steam produced by their own action, they may be set closer than in those boilers—as many marine boilers—in which they lie above a crown-sheet from

LAP-WELDED CHARCOAL-IRON BOILER-TUBES.

STANDARD DIMENSIONS.

Diameter—external.	Diameter—internal.	Thickness.	Wire Gauge.	Circumference—external.	Circumference—internal.	Transverse areas—external.	Transverse areas—internal.	Transverse areas—metal.	Length per sq. ft. of surface—external.	Length per sq. ft. of surface—internal.	Weight per foot.
In.	In.	In.	No.	In.	In.	Sq. In.	Sq. In.	Sq. In.	Feet.	Feet.	Lbs.
1.	.86	.072	15	3.14	2.69	.78	.57	.21	3.82	4.46	.71
1.125	.98	.072	15	3.53	3.08	.99	.76	.24	3.39	3.89	.8
1.25	1.11	.072	15	3.93	3.47	1.23	.96	.27	3.06	3.45	.89
1.32	1.15	.083	14	4.12	3.6	1.35	1.03	.32	2.91	3.33	1.08
1.375	1.21	.083	14	4.32	3.8	1.48	1.15	.34	2.78	3.16	1.13
1.5	1.33	.083	14	4.71	4.19	1.77	1.4	.37	2.55	2.86	1.24
1.625	1.43	.095	13	5.1	4.51	2.07	1.62	.46	2.35	2.66	1.53
1.75	1.56	.095	13	5.5	4.9	2.4	1.91	.49	2.18	2.45	1.66
1.875	1.68	.095	13	5.89	5.29	2.76	2.23	.53	2.04	2.27	1.78
2.	1.81	.095	13	6.28	5.69	3.14	2.57	.57	1.91	2.11	1.91
2.125	1.93	.095	13	6.68	6.08	3.55	2.94	.61	1.8	1.97	2.04
2.25	2.06	.095	13	7.07	6.47	3.98	3.33	.64	1.7	1.85	2.16
2.375	2.16	.109	12	7.46	6.78	4.43	3.65	.78	1.61	1.77	2.61
2.5	2.28	.109	12	7.85	7.17	4.91	4.09	.82	1.53	1.67	2.75
2.75	2.53	.109	12	8.64	7.95	5.94	5.03	.9	1.39	1.51	3.04
2.875	2.66	.109	12	9.03	8.35	6.49	5.54	.95	1.33	1.44	3.18
3.	2.78	.109	12	9.42	8.74	7.07	6.08	.99	1.27	1.37	3.33
3.25	3.01	.12	11	10.21	9.46	8.3	7.12	1.18	1.17	1.26	3.96
3.5	3.26	.12	11	11.	10.24	9.62	8.35	1.27	1.09	1.17	4.28
3.75	3.51	.12	11	11.78	11.03	11.04	9.68	1.37	1.02	1.09	4.6
4.	3.73	.134	10	12.57	11.72	12.57	10.94	1.63	.95	1.02	5.47
4.25	3.98	.134	10	13.35	12.51	14.19	12.45	1.73	.9	.96	5.82
4.5	4.28	.134	10	14.14	13.20	15.9	14.07	1.84	.85	.9	6.17
4.75	4.48	.134	10	14.92	14.08	17.72	15.78	1.94	.8	.85	6.53
5.	4.7	.148	9	15.71	14.78	19.63	17.38	2.26	.76	.81	7.58
5.25	4.95	.148	9	16.49	15.56	21.65	19.27	2.37	.73	.77	7.97
5.5	5.2	.148	9	17.28	16.35	23.76	21.27	2.49	.7	.73	8.36
6.	5.67	.165	8	18.85	17.81	28.27	25.25	3.02	.64	.67	10.16
7.	6.67	.165	8	21.99	20.95	38.48	34.94	3.54	.55	.57	11.9
8.	7.67	.165	8	25.13	24.1	50.27	46.2	4.06	.48	.50	13.65
9.	8.64	.18	7	28.27	27.14	63.62	58.63	4.99	.42	.44	16.76
10.	9.59	.203	6	31.42	30.14	78.54	72.29	6.25	.38	.4	20.99
11.	10.56	.22	5	34.56	33.17	95.03	87.58	7.45	.35	.36	25.03
12.	11.54	.229	4.5	37.7	36.26	113.1	104.63	8.47	.32	.33	28.46
13.	12.52	.238	4	40.84	39.34	132.73	123.19	9.54	.29	.3	32.06
14.	13.5	.248	3.5	43.98	42.42	153.94	143.22	10.71	.27	.28	36.
15.	14.48	.259	3	47.12	45.5	176.71	164.72	11.99	.25	.26	40.3
16.	15.43	.284	2	50.26	48.48	201.06	187.04	14.02	.24	.25	47.11
17.	16.4	.3	1	53.41	51.52	226.98	211.24	15.74	.22	.23	52.89
18.	17.32	.34	0	56.55	54.41	254.47	235.61	18.86	.21	.22	63.32

which enormous quantities of steam are liberated ; which steam, as well as that made by the tubes themselves, must traverse the intermediate spaces. Where the circulation is forced and rapid, the tubes may also be crowded more than where natural and sluggish. In locomotive boilers, the tubes, which are ordinarily from 1¾ to 2 inches in diameter, are set apart from one third to one fifth their diameters ; but the larger space is probably none too great.

43. The Details of the Problem, as coming to the designer and the constructor of the steam-boiler, are so largely matters determined by experience rather than by any scientific system or calculation, that much thought must be given to their consideration from the point of view of the practitioner in engineering and of the artisan engaged in building such structures,—from the boiler-maker's side rather than from that of the man of science.*

The selection of the iron or steel for shell, for stays, or of the rivets; the choice of style of riveting; the determination of the character of seam and lap; the decision of the question whether the use of reinforced seams or of heavier plates is likely to prove best in the end; the choice of type of boiler even, in view of known peculiarities of location or other conditions,—these must all be settled in conference with the boiler-maker, even if not directed absolutely by him. It seldom happens that the engineer making the designs feels competent to act throughout without consultation with his lieutenants in the workshop.

44. The General Considerations determining the design of a steam-boiler are, mainly, the following:

(1) It must supply a defined quantity of steam in a specified unit of time, or it must have a certain power.

(2) It must be as absolutely safe as it is practicable to make it.

(3) It must have reasonably high efficiency, and must be capable of working at the lowest total expense for fuel, attendance, interest on first cost, taxes, insurance, and all other running expenses, in proportion to work done, that may be attainable.

(4) It must be well suited to the location, and to all the special conditions affecting it when in operation.

Marine steam-boilers must, for example, be given the minimum practicable weight and volume, since it costs as much to

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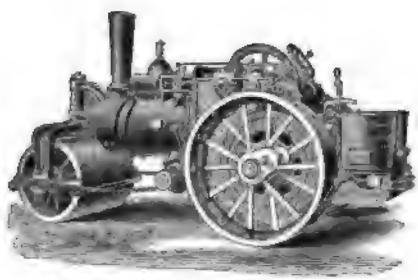
* See Thurston's Manual of the Steam-boiler, and, for practical details, Wilson on the Steam-boiler, edited by Mr. J. J. Flather.

carry a ton of boiler as a ton of cargo, and every cubic foot occupied by boilers, fuel, or machinery displaces a cubic foot of paying load. Naval boilers, also, must usually be kept as low in the ship as possible, to reduce risk of injury by shot. So important are these elements in naval construction that the practical limits of space and power on shipboard are commonly fixed by the space occupied by boilers; and the reduction of grate-area is the first problem attacked by the naval architect and engineer seeking high speed, whether for yachts, torpedo-boats, or larger craft. The proportions of the construction, generally, are discussed in Chap. VIII.



CHAPTER II.

VALVES AND GEARING; STEAM-DISTRIBUTION.

45. **The Office of Valves and their Gearing** is the proper distribution of steam to the engine in a manner determined by conditions assigned in advance. A perfect system would admit steam at full boiler-pressure up to the desired point of cut-off; it would suppress the entering current instantly and completely at that point, and would insure unimpeded expansion down to the desired minimum pressure at the end of the stroke; it would next open the exhaust-port instantaneously and fully, permitting an immediate fall of pressure to that of the condenser or of the atmosphere, and free outflow of the exhaust-steam at that pressure. Finally, the port would be closed in such manner as to secure, as nearly as possible, adiabatic compression to boiler, or other desired pressure at the end of the return-stroke. The ideal diagram of energy obtained by the steam-engine indicator, in this case, would thus be one having a straight and perfectly horizontal admission-line, a smoothly hyperbolic, or nearly hyperbolic, expansion-line, a vertical exhaust-line, a straight, horizontal back-pressure-line, and a compression-line of similar character to the expansion-line, $ABCIKA$, as in the next figure. This form of diagram is often approximated with slow-moving engines having a "detachable" valve-gear, but it is seldom seen otherwise.

In the accompanying sketch, in which the ideal and a modified form are compared, it is easy to trace some of the causes of difference.

At A the pressure of steam is usually a maximum. Should the induction occur at the right time and in the right way, the cylinder will be full of steam at the instant of forward movement of the piston; should the valve open late, A will be found

nearer B, and the line KA will be inclined toward the right; early opening of the induction-port will produce a line starting nearer M, and terminating at A as at first. If the pressure is not well sustained, AB will fall toward B; and if the cut-off does not take place promptly, the corner at B will be rounded off, as from H to G. At the end C of the expansion-line, similarly, early opening of exhaust will give QRS or PI; late opening may give CM, and the exhaust-line and back-pressure-line may become confounded. Early closing of the exhaust-valve may produce a compression-line MA. In all well-designed and properly constructed steam-engines, this compression, as

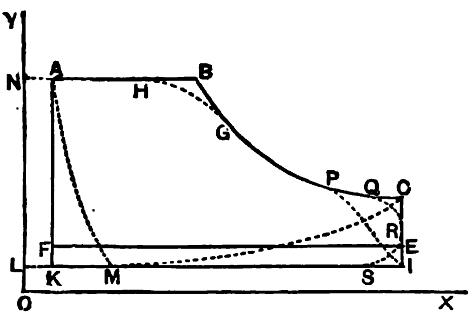


FIG. 64.—DIAGRAMS COMPARED.

well as the expansion, will be so arranged as to utilize to best advantage the available heat-energy of the fluid.

Some of these modifications of the ideal diagram are, therefore, due to practical conditions which dictate them. Thus, as the steam-ports are now made, in high-speed engines particularly, it is impossible to secure instantaneously, on opening them, the full pressure of steam in the cylinder; they are therefore given "lead,"—opened in advance. The same cause usually retards the inflow of the steam up to the point of cut-off, and thus produces a fall of pressure along the steam-line. Similarly, to meet the disadvantages inherent in the inertia of the fluid, as well as that of practically limited port-area, pre-release of the exhaust-steam is customary. The slower the

action of the expansion-valves and of the exhaust-valves, the more are the sharp corners of the ideal diagram rounded off.

The real diagram of the ordinary engine is of different form. The next figure illustrates these differences.

The admission-line *AB* is produced by steam on admission. Its normal direction is vertical, or nearly so, as it is traced while the crank is passing its dead-centres. Leaning outward indicates lead. With no lead it would lean inwards, as sometimes with condensing engines.

The steam-line *BC* is traced after the piston has commenced its stroke. Its proper direction is horizonal at a pressure nearly equal to that in the boiler; but this can only be ap-

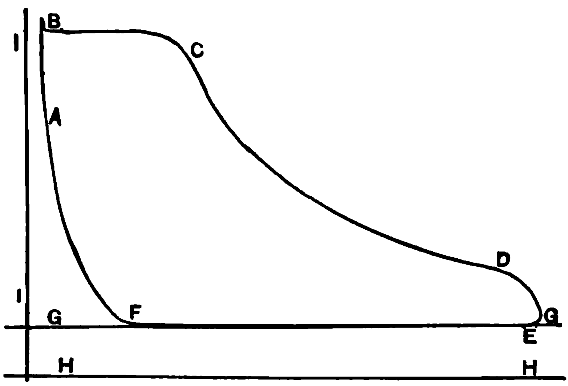


FIG. 65.—REAL DIAGRAMS.

proximated with such openings that the maximum velocity of flow will not exceed about 100 feet per second. But with throttling-engine diagrams the steam-line inclines downwards.

The point of cut-off *C* is the point where the entering steam is cut off. It is usually anticipated by a fall of pressure, which is less as the valve closes more promptly. With some engines, having multiported gridiron valves with detachable valve-gear, this fall of pressure is not appreciable, the point of cut-off is well defined.

When the instrument has been in good working order, the cut-off may be located at the point of contrary flexure.

The expansion-line *CD* begins at cut-off and terminates at exhaust.

The point of exhaust *D* is where exhaust begins; the expansion curve there ends, and the pressure begins to fall.

The exhaust-line *DE* is traced while the steam is escaping. When it occupies a considerable part of the return stroke, or nearly all, it indicates a cramped exhaust opening.

The back-pressure-line *EF* represents the pressure in the cylinder during the return stroke. With non-condensing engines the position of this line is somewhat above atmospheric pressure. With condensing engines it indicates a pressure somewhat in excess of that in the condenser.

The point of exhaust-closure *F* is anticipated by a rise of pressure; the eye may locate it very exactly.

The compression-curve *FA* exhibits the method and extent of variation of pressure after the exhaust-valve closes.

The atmospheric line *GG* locates the position of equilibrium of the piston of the indicator before steam is introduced.

The vacuum-line *HH* is parallel with the atmospheric line at such a distance below it as will measure the pressure of the atmosphere.

Port areas and the sizes of steam- and exhaust-pipe should be such as will permit the passage of the steam without serious loss of pressure. A velocity in pipes and ports not exceeding 100 feet per second, 6000 feet per minute, is usually considered satisfactory for the steam-pipe, and the steam-port should not be of less section. The Author has been accustomed to make exhaust-ports one fourth larger than the steam-port, since the steam must be in large part discharged in a very small fraction of a revolution of the engine, in order that the terminal may promptly fall to back-pressure. The exhaust-pipe should be at least as large as the exhaust-port, and some designers make it of considerably larger section. With high speeds of piston, the designer sometimes adopts higher velocities of steam and smaller sizes of pipe than suggested above.

46. The Varieties of Valve-gear in use are numerous and of very various types and forms. Where simplicity and

small cost have been the controlling considerations, the mechanism of steam-distribution is composed of few parts, but it is commonly unsatisfactory in its operation; where the prime requisite has been a good distribution and effective regulation, very considerable complication has often resulted. Of these two cases, the common three-ported valve, driven by a single eccentric, as in the locomotive or the cheaper styles of stationary engines in the market, illustrates the first, and the universally familiar Corliss, or the Greene, engine exhibits the second; while there are numerous varieties of all intermediate degrees of complexity. The precise character of the system adopted in any case is determined partly by the form of valve to be used, as well as by the nature of the general problem presented for solution. In some cases it is merely required that a fairly good steam-distribution shall be secured; in other instances a good method of expansion must be obtained; in still other cases the system must be capable of adjustment with a view to nice regulation of speed; and in all locomotive and marine engines easy and prompt reversal of the direction of motion is as essential as any other requirement. The engine with single three-ported slide-valve, single eccentric, and throttling governor; the engine having a separate cut-off valve on the back of a three-ported slide; the modern "automatic" high-speed engine, and the various detent or detachable valve-motions: altogether exhibit such a series of forms of varying complexity and efficiency.

In all forms of "detachable valve-gear," or "drop cut-off," two trains of mechanism are employed: the one receives motion from the eccentric or other appropriate part of the engine, and transmits the movement to a convenient location on the machine where it operates an "active" catch, latch, or mechanical equivalent; the other train is connected with the steam-valve at its remote extremity, and is so related to the first that the "passive" catch which forms the other terminal element may be, at pleasure, thrown into or out of gear with the "active" element terminating the first train. The two catches being in contact, the two trains become one, and

the motion of the valve is controlled by the eccentric until, at a fixed instant, detachment takes place, when the valve is closed by independent means, as by a weight, a spring, or by steam-pressure.

Many varieties of detent gear have been devised; especially in making variations upon the Corliss gear. In some cases the detachment is effected by a movement of the part acting as a driver; in others the driven piece is given the movement causing separation. It is a matter of no importance, kinematically, where, in the train, the detachment occurs; but it is usually best, for obvious reasons, to effect it as near the valve as is possible. The weight and inertia of the detached parts being thus made a minimum, the closing motion is rendered quicker, and it is easier to prevent slam and jar at the instant of arrest of motion.

47. **The Classification of Valves and Regulating Gearing**, as already seen, involves the consideration of the form of valve, method of steam-distribution, system of regulation, and such special functions as reversal of the engine.

Valves are constructed in an enormous variety of forms, adapted to the various kinds of engine and to the methods of opening and closing ports adopted by designers. The most simple and common is the three-ported slide, used almost exclusively on locomotives. In some cases this is supplemented by a slide cut-off valve riding either on its back or in a separate chest. Often a two-ported valve is placed at each end of the cylinder, the two valves being worked together as one, of the preceding form; in still other cases a valve is placed at each end for induction, and a similar pair is used for eduction. All these forms are determined by the system of distribution. In some cases the valve is a flat, sliding plate; in others it is a rotating piece with cylindrical, conical, or other surfaces of revolution, as bearing-surfaces, in still other arrangements several ports are used, and a "gridiron" valve is made to cover or to uncover them all, simultaneously. In a few engines continuously rotating valves have been employed.

Valve-gearing includes the whole system: valve, driving

mechanism, cut-off gear, regulating apparatus, and reversing gear. The driving mechanism consists, usually, of one or more eccentrics, their rods, and their connections with the valve-stems. In some cases the eccentric is displaced by other contrivances. The substitutes are various sorts of cams, and links "paired" with the connecting-rod. The cut-off gear is sometimes arranged to control and detach the main steam-valves, sometimes to operate an independent, or cut-off, valve, often to adjust the eccentric.

The regulator is usually the "fly-ball" governor of Watt, and is so attached as either to control the steam supply entering from the steam-pipe, by adjusting a "throttle-valve," or to fix the point of cut-off by acting upon the cut-off valve.

The reversing-gear is commonly that seen in the locomotive, the "link-motion;" but it is occasionally of some more ingenious but usually less simple form.

The varieties of valves to be here studied are:

(1) The three-ported valve.
(2) The two-ported valve.
(3) Detached or single-port valves.
  (*a*) The slide.
  (*b*) The Green valve.
  (*c*) The Corliss valve.
  (*d*) The Goodrum revolving valve.
  (*e*) Puppet-valves.
(4) Balanced valves.

The forms of valve-gearing to be examined are:

(1) The single eccentric and valve.
(2) The Meyer and related systems with two eccentrics.
(3) The Cornish system.
(4) The Stevens valve-gear.
(5) The Sickles cut-off.
(6) The Corliss type of valve-gear.
(7) The Green type of valve-motion.
(8) Cam-motions.

(9) The Porter-Allen gear.
(10) Automatic gear with shaft-governor.
(11) Continuously rotating valve-motions.

The reversing-gears to be considered are:

(1) Stephenson link and reversing lever.
(2) Gooch's link.
(3) The Allan motion.
(4) Walschaert's and Strong's gears.
(5) Loose rotating eccentrics, shifting eccentrics.
(6) Brown's and Marshall's gears.
(7) Joy's gear.
(8) Steam reversing-gears.

A Correct Steam-distribution, as has been elsewhere shown, is one which gives as nearly the Carnot form of cycle as is practicable; taking into the engine steam at as nearly boiler pressure and temperature as is possible, retaining it to the point of cut-off; expanding as nearly adiabatically as circumstances permit; exhausting at a constant minimum temperature and pressure; and finally compressing steam into the clearance and port-space and to as nearly boiler-pressure as is practicable. These conditions are only attainable when the engine is precisely adjusted to its work by correct proportioning and by proper choice of the ratio of expansion; the best value of which should be ascertained by reference to the financial elements of the case, and by methods elsewhere detailed.

The best form of valve-gear is evidently that which, simple and inexpensive as possible, permits such a steam-distribution as is above described. But variations perpetually recur in the work demanded, and in the speed and the effort of the engine, to meeting which variations an automatic system of self-adjustment is usually demanded; and this indicates another essential condition of the successful operation of any valve-gear. The Author has suggested that a simultaneous variation of the ratios of expansion and of compression may sometimes be desirable, such as is illustrated in the operation of the Stephen-

son link; and that the best among existing valve-gears, for any special case, may prove to be that which, combining a variable expansion with a variable compression, is also capable of prompt and exact adjustment by a sensitive and efficient governor.*

48. The Three-ported Valve, and its action, as illustrated in nearly all the simpler and less expensive engines in use, combines, in the simplest and most efficient manner yet devised, provisions for supplying steam alternately to each side of the piston, and for conducting the exhaust steam away from the opposite side and discharging it from the cylinder. It, at the same time, affords the most convenient possible means of obtaining a motion, in the opening and closing of ports, such as will give an area of free passage varying so as to give a satisfactory distribution of steam. Ordinarily it is subject to considerable friction; but many ingenious and successful plans for balancing it are in use, and its friction may be thus rendered unimportant. Its simplicity of form, ease of construction, and general efficiency have made it the most common of all known steam-distribution valves. It was invented by Murray in 1799.

The form of this valve, which is often called the short-D, or locomotive, slide, is seen in section in the figure. *AB* and *CD* are the two working faces, bearing on the valve-seat and cutting off the access of steam to the inside of the valve and the ports covered by it. *E* and *F* are the cylinder steam-ports, through which steam is alternately admitted and exhausted, at either end, as the piston, *G*, traverses forward and backward between *H* and *I*. The shell of the valve, *J*, is not only made heavy enough to bear safely the pressure of steam upon it and to resist any tendency to spring, but also sufficiently strong to take the push

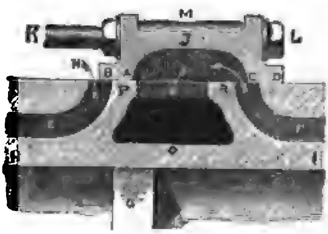


FIG. 66.—THREE-PORTED VALVE.

* Expansion of Steam and Regulation of the Engine; Trans. Am. Soc. Mech. Engrs., 1881; Jour. Franklin Inst., Oct. 1881.

and pull of the valve-stem, *KL*, which is either held by nuts, as shown at *M*, or oftener, in locomotive practice, by a stirrup or strap carried around it in the horizontal plane. The steam-chest, *M*, encloses the valve. The action of the valve is as follows: The piston, *G*, moving in the direction of the arrow, steam enters at *N*, passing through the port *E* to the end of the cylinder, *H*, and, there entering, drives the piston toward *I*. When the piston passes the middle point of its stroke, the valve, which had previously been moving toward the right, reverses its motion. At this instant the port *E* is wide open, as is the exhaust-port *F*; but *E* is open to steam and *F* to the exhaust at *C*, steam moving into *E* and out of *F* as shown by the arrows. This reversal occurring, the ports are gradually closed, as the piston moves more and more slowly; and when the latter stops at *I*, the crank turning the centre, both steam and exhaust-ports are closed; but, if *AB* and *CD* are made of the same width as the ports *E* and *F*, the continued movement of the valve immediately opens the port *E* to the exhaust and the port *F* to the steam, and the piston is driven back with precisely the same motion seen in its forward movement; the steam entering at *CD*, and the exhaust passing out at *AB*, and over *P* into the outlet at *O*.

But it is generally desirable that steam should not "follow" full-stroke as just described; it is found vastly more economical to suppress the induction at half-stroke, or even less if possible. It is also advisable, especially with engines of high speed, to give "*lead*" to the valve, i.e., to so adjust it that steam may enter the cylinder, and that exhaust may take place, a little in advance of the beginning and end of the stroke. These results are accomplished by the "*lap*" and *lead* of the valve, and by proper modification of its motion. By *lead* is meant the extent to which the valve, if without lap, is in advance of its middle position at the end of the piston-stroke; in other cases, it is the extent of port-opening at that instant. By the *lap* is meant the extension of the valve at either end, at *B* or at *D*, so as to cause it to cover the port a longer time and during a larger proportion of the period of motion of the

piston than when not thus built out. The greater the lap, the longer the time of covering the port and of, consequently, suppressing the steam during a portion of the stroke which is the greater as the lap is greater. The effect of lead is also to make not only the steam- and exhaust-opening take place earlier, but also to produce an earlier suppression of the entering steam. By thus giving lap and lead to the valve, any point of cut-off and ratio of expansion up to half-stroke can be readily attained, and, by permitting considerable compression or "cushioning"—a consequence of lead of the exhaust—even more expansion may be obtained.

The *nomenclature* of the case is the following: Assume the valve at the middle of its path; both steam-ports being covered. If there be lap, the parts *AB* and *CD* being wider than the ports, the amount of excess on the outer edge *ab*, and the extent by which the valve overlaps the port, is a measure of the *outside lap*. Similarly, the cover *cd*, on the exhaust side, measures the *inside lap*. The valve being in operation, the extent by which the steam-port is opened when the crank is "on the centre," and the piston just beginning its stroke, is the *lead*.

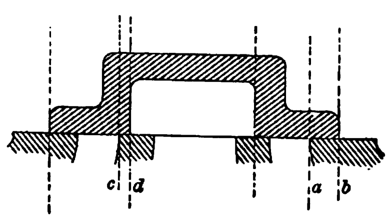


FIG. 67—LAP OF THE VALVE.

The method of construction of this valve, *B*, and of distribution of steam by its operation, are shown in the accompanying figures. The first of the pair shows the steam entering at the top, *de*, and the exhaust, *gf*, from the bottom of the cylinder, the piston, *K*, moving downward. In the second, the position of the valve is reversed, the piston moving upward and taking steam below, while the exhaust passes out at the top. Steam enters the steam-chest, *CE*, at *D*, and the exhaust through *d* passes out at *O*. *F* is the valve-stem, and *S* the piston-rod sliding, steam-tight, through the stuffing-box.

Where, as is usual, the valve is given lap, the eccentric must be given correspondingly increased throw. Of two valves

working over the same size of port, the opening demanded being the same in each, that having lap will necessarily be given an increased movement by that extent. The same principles, as is now seen, apply where exhaust-lap is given. Since lap on either steam or exhaust delays opening the port, and similarly hastens its closing, the eccentric must be given in either case a corresponding advance to insure the retention of the desired lead. The opening of the port is always equal to

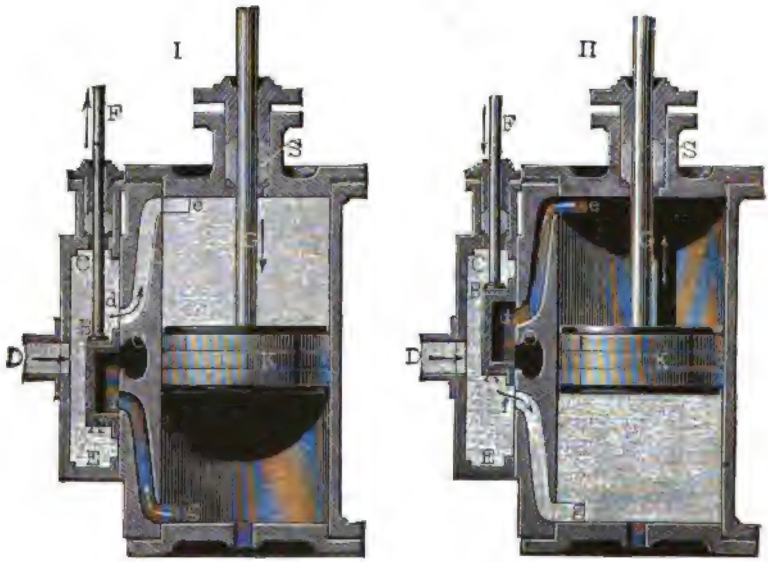


FIG. 68. THREE-PORTED VALVE. FIG. 69.

the throw of the eccentric diminished by the lap. With the three-ported valve, the exhaust-lap is generally made small or *nil*, and the exhaust-opening is, or should be, greater than the steam-opening.

Lap is measured on the valve when at mid-travel; lead is measured on the port when the crank is on the centre and the piston at the end of stroke.

The point of cut-off by the plain slide-valve is

$$\frac{1}{r} = 1 - \left(\frac{2l}{s}\right)^2,$$

where r is the ratio of expansion, and where l is the lap and s the stroke or travel of valve in the same units. Thus, let

$$l = 1''; \quad s = 5'';$$

then

$$\frac{1}{r} = 1 - \left(\frac{2 \times 1}{5}\right)^2 = 0.84; \quad r = 1.19.$$

Lead seldom exceeds $\frac{3}{8}$ inch, even in fast locomotives; though sometimes above one-half inch, and the angle of lead rarely attains 10°. Very large engines are seldom fitted with the slide-valve, except at sea, and in these the lead sometimes exceeds an inch, the angle of lead being restricted to similarly small dimensions. Lap has a vastly wider range, being often as much as 3 inches or more in marine engines and an inch or more in locomotives; inside lap, where introduced, having about one half the magnitude of outside lap. The stroke of the valve ranges from 4 or 5 inches on the locomotive to sometimes a foot or more in marine-engine slides.

Although, with direct connection, it is impossible to modify one element of slide-valve action without affecting every other, and thus it is impracticable to secure, at the same time, equal lead and equal expansion at both ends, a modification of the connecting link-work may often be made, which shall at least approximately produce any such desirable change; as is seen, for example, in Professor Sweet's designs.

With a proposed wide range of variation of cut-off, the slide-valve must often be given such considerable motion that it will exceed, sometimes greatly, that needed to fully open the port. This excess, or "over-travel," must be noted in proportioning the valve; otherwise the edge of the valve may pass that of the exhaust-opening and permit "blowing through." Over-travel gives increased quickness of opening and of cut-off, but requires more power. Where quick speed of engine and large steam-opening are demanded, double-ported valves are employed to give large opening and reduce excessive travel.

49. **Single and Double, and Multiple, or Gridiron, Valves** are all familiar forms to the engineer. A serious disadvantage of the valve just described is the length of port, and

consequent large volume of "dead-space" between the valve and the interior of the cylinder. All the steam entering and filling this space is at each stroke, in the absence of provision for cushioning, useless, or nearly so, in the production of work, and is thus wasted. This loss may be reduced by early closure of the exhaust-port and resulting compression; but it is not always practicable or advisable to submit to this inconvenience. The ports may be shortened by making a valve-seat at each end of

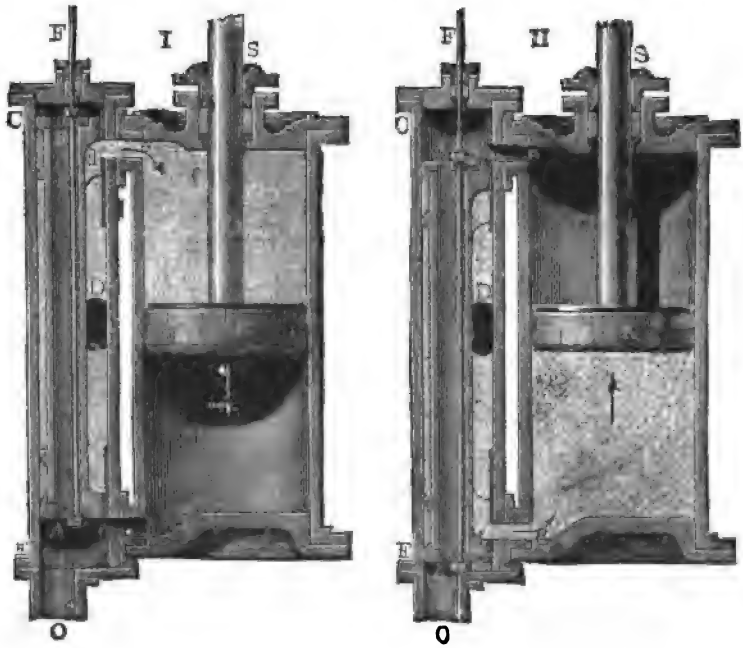


FIG. 70. LONG SLIDE. FIG. 71.

the cylinder, and lengthening out the valve so as to extend from end to end, as with the early form of valve known as the "long D"; but a better plan is to make two valves, one for each end, each having the shape and proportions of the corresponding half of the three-ported valve. Each valve is thus a two-ported valve, its lap covering the steam-port at its own end of the cylinder, and its interior communicating by an independent port with the condenser or exhaust-pipe.

The longer style of valve is seen at *AB* in the next pair of

illustrations; the steam entering at D, passing under the valve, surrounding it, and passing into the upper end, in the first case, at d. The exhaust takes place at A, past the end of the valve, and out by O. The reverse stroke is effected as shown in the second figure, steam entering at g, and the exhaust leaving at e, and traversing the valve from end to end and out at O. One objectionable feature of this device is that, the entering and the exhaust steam exchanging heat over a large surface, a waste is occasioned which goes far to annul the gain by the shortening of the exhaust-port.

The gridiron valve is usually fitted to a port at each end of the cylinder. It consists of a series of transverse bars and passages alternating, covering a similar arrangement in the seat, all the passages in the seat communicating with the port at that end of the cylinder. This permits the opening of a large port with but a slight movement of the valve, giving a decided advantage in the direction of reduction of friction and of waste of power.

FIG. 72.—DOUBLE-PORT D VALVE.

A double three-ported valve is sometimes used to secure a good distribution in the compound engine, as seen in the figure; where steam passes from the chest through $\ddot{u}$ to act on the upper side of the small piston in WW, and its exhaust goes through u, k, $\ddot{U}$ to act on the larger piston in the other cylinder.

Reversing the position of the valve, the steam enters the small cylinder at u, and leaves by $\ddot{u}$, k, U.

A total area in any case, for steam-ports, equal to

$$a = \frac{Av}{6000},$$

where A is the area of the piston and v its velocity in feet per minute, is sufficient for the exhaust-port; and for the steam-ports one fifth less will suffice.

The Ide "double-admission" valve is a piston-valve, seen in Fig. 73, and exhibits the general plan of the preceding as

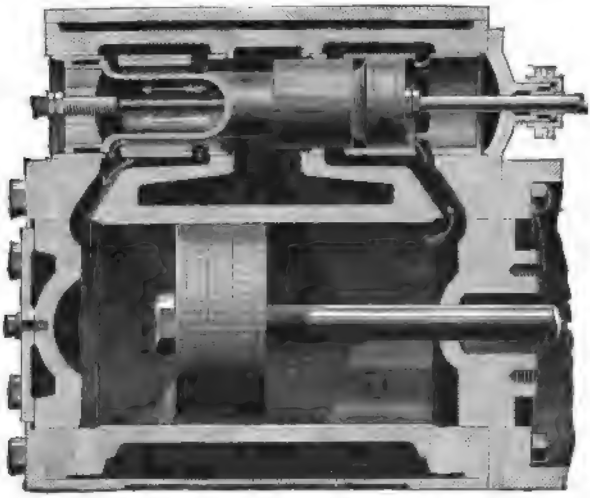


FIG. 73.—DOUBLE-PORT PISTON-VALVE.

applied to this class of valve. The double-ported valve finds special application in engines of high speed, and is assuming constantly greater and greater importance.

A peculiar modification of the gridiron valve is that de-

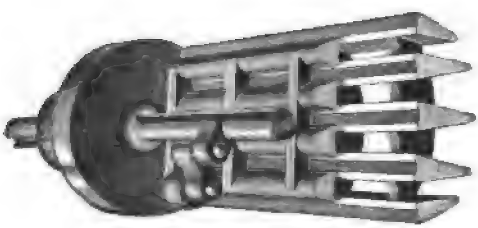


FIG. 74.—THE WHEELOCK VALVE.

signed by Mr. Wheelock for the later engines of that class. It is, as seen in the illustration, a valve and seat set together in a slightly coned, nearly cylindrical, plug, so proportioned as to

give a large port-opening, and so arranged as to be readily inserted or removed in case of repair, or where it is desired to inspect either valve or seat.

50. Later Valves of the class last described have been given many different forms by various recent designers and builders of engines. The three-ported valve is in very general use where high efficiency is not aimed at; the long slide is now and then employed, but not frequently; the gridiron valve, excellent as it is for its purpose, is even less generally used; while the cylindrical and conical valves, rotating about an axis, have continually become more common.

The Greene valve was formerly a perfectly plain flat valve,

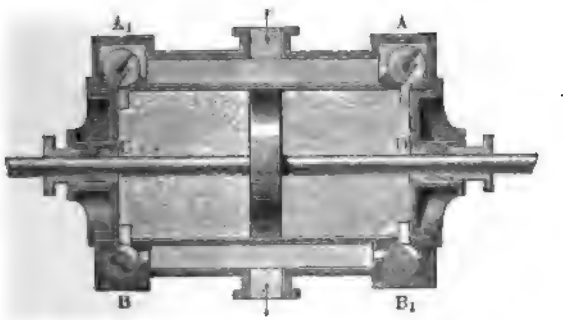


FIG. 75.—CORLISS VALVE.

one employed for steam and another for exhaust at each end of the cylinder.* Later Greene engines have often been fitted with gridiron valves.

The Corliss valve, seen in section in the figure, has a cylindrical face and seat, and is rotated by a spindle loosely fitted to it, which spindle is operated by a rocker-arm outside the steam-chest. The method of steam-distribution is precisely the same as with the plain single-ported valve. The separation of the steam from the exhaust-ports in this engine is productive of economy, not only by shortening the steam-passages, but, also, by reducing the waste of heat by the variation of

* Thurston: History of the Steam-engine.

temperature due to the passing of alternately cold and hot currents of steam over the same surfaces.

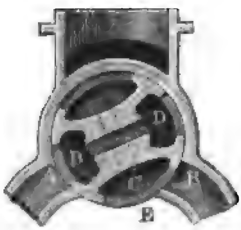


FIG. 76.—PLUG-VALVE.

An older form of rotating valve is the three-ported valve seen in the next figure. It, in fact, antedates the simple three-ported valve of Murray. The steam, passing through *DD*, enters alternately by *A* and *B*, while the exhaust-steam escapes through *C*. The channel *DD* balances the valve.

51. Poppet-valves, or "puppet-valves," so called because they rise and fall, instead of sliding on their seats, are often used on vertical cylinders and occasionally on horizontal engines. They are less used than formerly, however.

The single poppet-valve consists of a disk fitting the orifice of a pipe or a circular opening in the valve-casing, and raised and depressed by means of a stem which is secured at one end

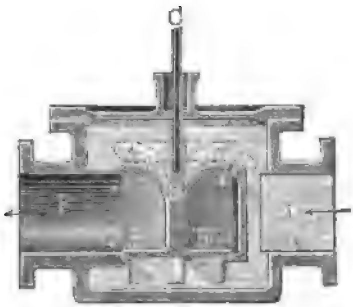


FIG. 77.—DOUBLE POPPET-VALVE.

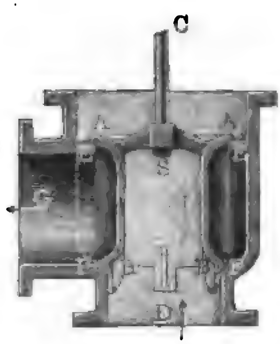


FIG. 78.—REULEAUX VALVE.

to its centre, and, passing out through a stuffing-box, is attached to the proper detail of the general system of valve-gearing. Such a valve, however, sustaining full steam-pressure on its whole area, is difficult to lift, and, when of large size, must be balanced. This balancing is effected by using two disks of nearly equal size (Fig. 77) carried on a single stem. Here *AA* is one and *BB* the other disk; *SC* is the stem. Steam entering at *D* surrounds both valves and their seats. Many modi-

fications of this valve are in use, several of which are here shown. That in the next illustration is the Reuleaux valve, in which the valve is a section, *AB*, of pipe fitted to seats at either end, *AE*, *FF*, the steam-pressure being fully balanced at all times.

The "equilibrium valve," or Cornish valve of Harvey and West, is what may be termed an inverted poppet-valve. The valve is a pipe or box, *CC*, seating on a cage, *AB*, at *aa*, *bb*, and *cc*, *dd*. When open, as in the next figure, steam, entering from *D*, follows the arrows, and, the valve remaining balanced, its

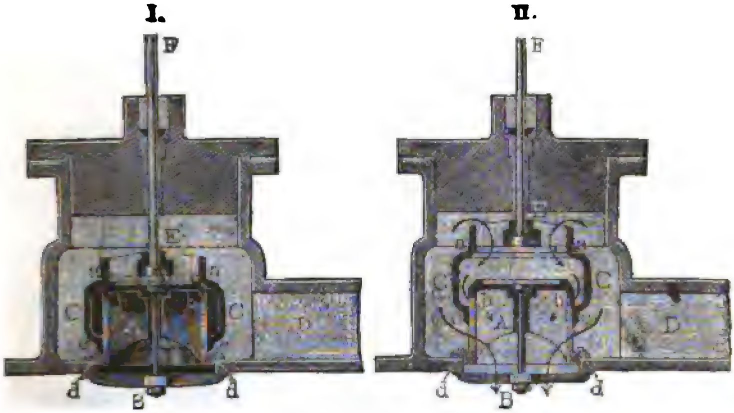


FIG. 79. CORNISH VALVE. FIG. 80.

manipulation is comparatively easy. This form of valve has been very extensively used in the Cornish pumping-engine.

Fig. 81 is such an arrangement as constructed in 1855 for the U. S. steam frigate Wabash.

C is the steam-valve; *DD*, the cut-off valves, attached to the valve-stem *E*; *FF*, rings in the steam-chest cover for balancing the valve.

To alter the ratio of expansion, a hand-wheel on the end of the valve-stem *E*, turned in one direction, will draw the blocks together; and to cut off shorter, the action of the screw is reversed.

The article on Valve-gearing may be referred to for more of detail and for principles of operation.

52. Balanced Valves are essential in some forms of steam-engine, and are very desirable in all cases, especially where the pressure is great and the valves large. Examples of balanced valves are given elsewhere. As a matter of economy, balancing

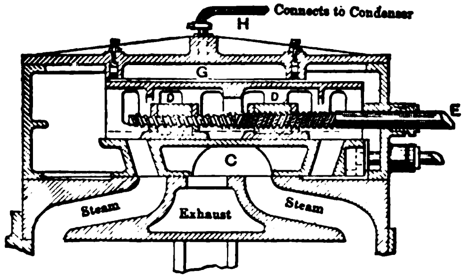


FIG. 81.—MEYER VALVE.

is most important with the several forms of slide-valve. The single poppet-valve is difficult to raise from its seat; but the expenditure of the total work is not large in that direction. On the other hand, the slide-valve, pressed upon its seat by the steam, offers considerable resistance to movement, and absorbs a somewhat large fraction of the power of the engine;

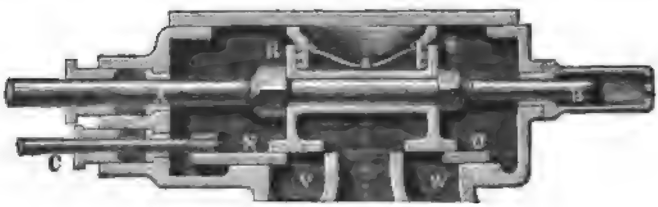


FIG. 82.—BALANCED VALVE.

to say nothing of the further disadvantage of, in many cases, reducing the efficiency of the regulating mechanism.

The method of balancing a slide-valve is commonly introducing a seat above, symmetrical with that below the valve, and opening the latter at the back, thus securing an equilibrium; little or no pressure reaching the valve, except on its sides and ends, and the whole being thus placed in balance. This is seen in the figure, in which a ring, *RS*, mounted on the back of the valve, *AB*, and bearing on the inner surface of the bonnet,

held in place by the spring shown within it, preserves nearly an equilibrium of vertical forces.

The piston-valve, of which an illustration is here seen, is a form of balanced valve which has come into extensive use on account of its comparatively small cost. When the water used

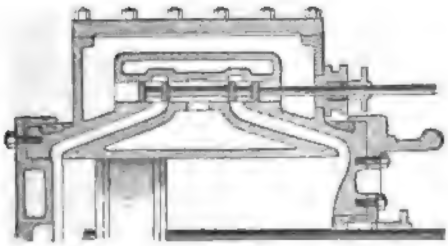


FIG. 83.—PISTON-VALVE.

is clean, and not liable to carry gritty matters into the engine by "priming," this valve works well, and it has come into use on even the largest of marine engines. Other illustrations of balanced valves will be shown in connection with descriptions of various forms of valve-gear.

The Friction of the Slide-valve, when unbalanced, and that of its gearing, constitutes a very considerable fraction of the total friction of the engine. Thus, experiments conducted by the Author have shown a good form of slide-valve on a small engine to produce about one fourth the total friction of the engine; while the balanced valve only exhibited from one fifth to one tenth as much, or from five to two and a half per cent. The balancing of the valve is thus seen to be a somewhat important matter in itself, as well as essential to the satisfactory action of the governor.

In many cases the friction of the unbalanced valve is very much greater than in the cases just referred to. An ordinary locomotive valve often carries a pressure of eight to ten tons, and has a frictional resistance to sliding amounting to from two to three tons at the valve-stem; and the power demanded to operate the valve amounts to twenty-five or thirty horse-power. The pressure on the bearing area of the valve is often

250 to 300 pounds on the square inch, and the coefficient of friction is apt to be very high.* Proper balancing reduces the pressure on the valve-seat so greatly that a very sensible economy in fuel may be secured by this means; while the expense of wear and repair is also lessened.

Fig. 84 shows the form of a balanced valve in which a "relief ring" is employed to take the pressure off its back. The space above the middle of the ring communicates with the condenser, in condensing engines. The diaphragm shown is of spring steel and holds the frame against the valve, on which it works steam-tight. The piston-valve, which it is always possible to make satisfactory in this respect, is the most usual type.

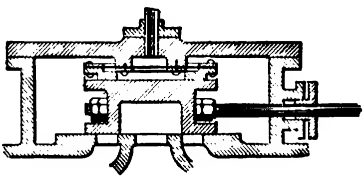


FIG. 84.—THE BALANCED VALVE · RELIEF-RING.

The use of balanced valves has become much more general since the introduction of high steam-pressures in consequence of the gain in reduced friction, but probably even more because of the difficulty arising in working unbalanced valves under such high pressures and their tendency to "cut" and to wear their faces and seats. These causes have combined to drive the unbalanced flat valves of earlier days from the marine engine, in which they are very largely replaced by piston-valves. The latter are, at sea, little likely to be injured by gritty matter brought over from impure water in the boiler, a common difficulty on land.

This valve, as used on stationary engines, particularly, is made in great variety of form and proportions. A typical example is that designed by Mr. Ide, here shown; and other illustrations of the same class of valve are to be seen in many of the engravings distributed through this work, exhibiting the construction of all kinds of engines.

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* Coefficients have been observed exceeding 10 per cent, which may be taken as a fair figure for designing. See Mr. Gidding's data; Trans. Am. Soc. M.E., vol. II.

The perfect balance of these valves, their simplicity and small cost of construction, make them a favorite with builders of high-speed and of marine engines.

In the case of the McIntosh and Seymour engine, the valve

FIG. 85.—PISTON-VALVE.

illustrates the construction of a good form of piston-valve. The valve is a cylinder, with cast-iron ends, a tube connecting the two ends. To prevent leakage, an "adjustable seat" is employed, shown separately in Fig. 86. It consists of a crescent-shaped ring having steam-ports through it which match the ports in the steam-chest. The ring is split and is adjusted by a stem which extends to the upper side of steam-chest, where it can be turned by a wrench. This seat rests in a recess formed by widening the steam-port all around the valve-opening, and is held in position by a cover-plate. This plate forms one side of the recess, and the opposite side of the widened port forms the other, the valve-seat fitting steam-tight

between them. If any wear occurs between the valve and seat, it can be compensated by turning the wrench.

The piston-valve and many other forms of balanced valve may be arranged to transpose steam and exhaust sides, the former taking steam either inside or outside, as may be found desirable.

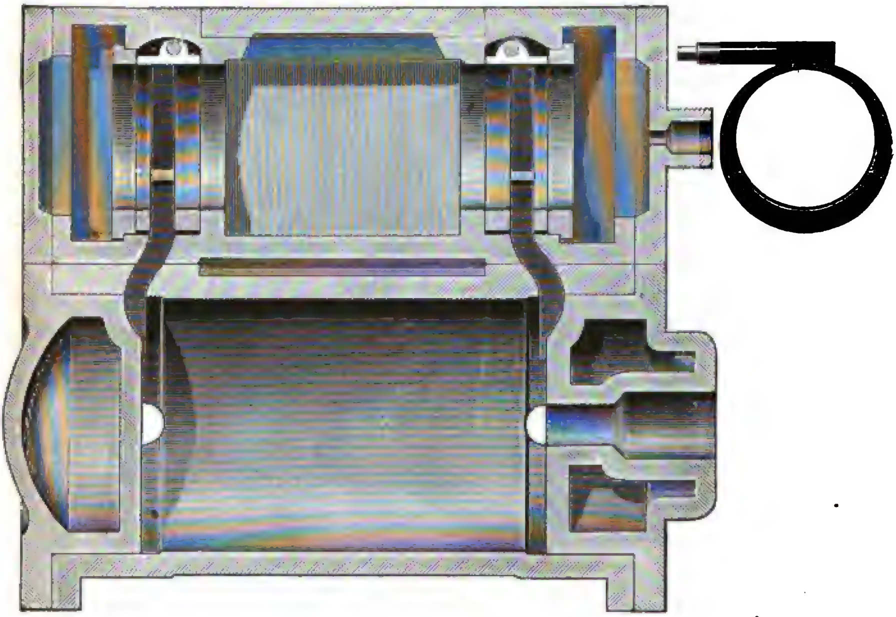

FIG. 86.—VALVE AND STEAM-CHEST.

53. The Single Valve and Eccentric may drive a flat locomotive-slide, or any equivalent, as the piston-valve; the motion is the same in either case. The maximum total movement of the valve is at least equal to double the sum of the width of port and the valve-lap; it is the greater, therefore, the greater the lap and the shorter the cut-off; but, as will be presently seen, this simple gear will not permit satisfactory distribution with a large ratio of expansion.

On account of the angularity of the eccentric- and connecting-rods in mid-throw, the relative motion of valve and piston is not the same on both sides the half-centre, or of the middle

of the piston-stroke; but the great length of the eccentric-rod, in comparison with the valve-movement, reduces its effect, in producing this lack of symmetry of action, to an unimportant amount. The effect of the connecting-rod is more serious; but with long rods it is often neglected, and the motion is assumed to be strictly harmonic.

Where the valve is given lead, it is effected by setting the eccentric ahead of its normal position, without lap or lead, by the amount required to cause it to open the port as much earlier as is desired.

The *angular advance* of the eccentric is the angle through which it is thus set ahead. The "radius of the eccentric," as the term is commonly employed, is the distance between the centre of its disk and that of the shaft on which it turns, and is equal to half the travel of the directly connected valve. The "throw" of the eccentric is usually taken to mean this radius, but is sometimes taken as the full valve-travel. The eccentric is, in fact, a crank having the sum of the lap and port-space as its radius, and its pin greater in diameter than the orbit of its own axis or centre.

The lead and the lap of the valve may be measured either at the valve or at the eccentric. The lead is equal, in linear measure, to the distance the port is opened at the instant of beginning the stroke of piston; or by its ratio to the half-travel of the valve, by what is called the "*ratio of lead*"; or it is measured by the lead-angle of the eccentric, that angle by which it is ahead of the position corresponding to the middle of the valve-stroke. Eccentrics set to give, respectively, forward and backward motion, without lead and with no lap on the valve, are directly opposite; but, if lead be required, each being set ahead, relatively to its own proper direction of motion, the angular distance, α, required to give that lead, they are then separated by an angle of $180° - 2\alpha$, a semicircumference less double the angle of lead.

Similarly the lap is measured either by its linear magnitude as taken on the valve; by the ratio of that quantity to the half-travel of valve, the "*ratio of lap*;" or by the angle which

the eccentric must be set ahead to take up the lap and still permit the port to be opened at the right moment; or, finally, by the *angle of lap*. In any case, the ratio of lead of the centre of the valve, or that of lap, is equal to the sine of the angle of lead or that of lap.

The action of the slide-valve of the three-ported variety, without lap or lead, is seen in the three figures here grouped together. In each the centre line *oy* indicates the middle of the travel of the valve and of the ports; *sv* designates the line of motion of the valve, *v*, and its stem, *s*; *R* is the eccentric-rod; *ef* measures the eccentricity of the eccentric; and *C* is

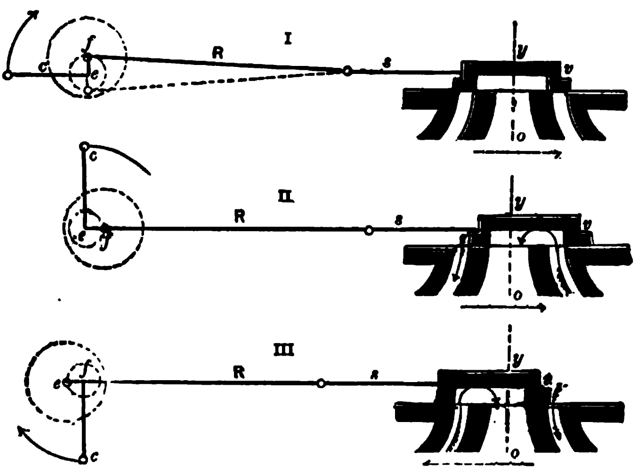

FIG. 87.—ACTION OF THE SLIDE-VALVE.

the crank. In I, the crank is "on the centre," the eccentric at right angles to it, and the valve just covers both steam-ports, neither admitting steam into the one end of the cylinder nor allowing exhaust to occur from the other end. In II, the crank and eccentric have swung through an arc of 90 degrees, the piston is near the middle of its stroke, and the ports are both wide open. In III, the crank and eccentric have described another half-circumference, and the piston is half-way back on the return-stroke, the ports having been, meantime,

closed and again fully opened. The engine will run one way as well as the other with this disposition of valve and eccentric.

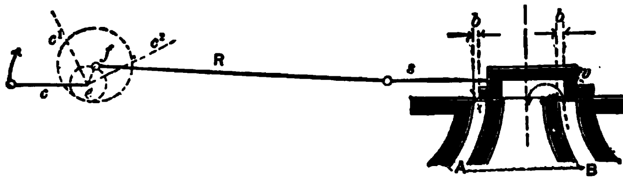

FIG. 88.—VALVE WITH LEAD.

In the next figure the effect of setting the eccentric ahead, its angular advance being *Cef*, is seen in the production of lead, or linear advance, to the extent, *bb*, opening both steam and exhaust ports alike. The direction, *A B*, of the motion of the piston now becomes definite, and is indicated by the arrow. With no lead and no lap, as in the preceding case, the engine will turn either way indifferently.

Exhaust-lap, or "eduction-lap," is often given when the lead becomes so great that the exhaust opening would otherwise take place too early, as, often, when the valve is set to give considerable expansion, cutting off, as is perfectly practicable, at one-half or five-eighths stroke.

In the illustration, Fig. 89 is a section of half a slide-valve and its port having considerable lap; *W* is the lower port; *X*, the lower half of the valve, in its middle position; *U* is the *steam side*, and *V* the *exhaust side*, of the port; *C* is the *induction edge*, and *P* the *exhaust edge*, of the valve; *UC* is the *lap on the steam side*, and $\overline{VP}$ the *lap on the exhaust side*.

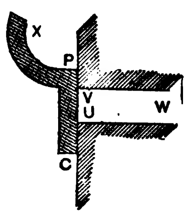

FIG. 89.—LAP OF VALVE.

Thus, studying the action of the simple slide-valve and eccentric, it is seen that, the motion of the crank and shaft being made uniform by the fly-wheel, the movement of the piston would be the parallel component of the motion of the crank-pin; but the varying angularity of the connecting-rod actually produces such a modification of this relative motion that the piston is always ahead of its otherwise normal position during the forward-stroke, and behind during the return-stroke. Neg-

lecting this, which is always a sensible effect, the valve without lap, driven by an eccentric without lead, holds the steam and exhaust ports partly or fully open throughout the stroke. When given lead by setting the eccentric ahead, the times of opening and closing of all the passages are correspondingly and equally advanced; and the addition of lap on the steam-side, with appropriate further advance of the eccentric, produces an early closure of the steam-port and expansion of the steam, while giving still more lead to the exhaust, opening and also closing that port in advance of its proper time. This *last* defect is corrected, if serious, by giving the valve *negative* exhaust-lap; otherwise, the more serious defect being early opening of the exhaust, lap may be *added* to the exhaust, to prevent too early discharge of the steam and premature cessation of expansion. By adding a little more lap on the crank-end of the valve, the irregularity of steam-distribution due to the angularity of the connecting-rod may be compensated.

The four characteristic periods in an engine-cycle being those of admission, expansion, exhaust, and compression, it is seen that, changing lap, lead, and travel, one at a time, the following effects follow:

(1) *Increased Outside Lap.*

(*a*) Admission is later; ceases sooner.
(*b*) Expansion occurs earlier; continues longer.
(*c*) Exhaust is unchanged.
(*d*) Compression begins at same point.

(2) *Increased Inside Lap.*

(*a*) Admission unchanged.
(*b*) Expansion begins as before; continues longer.
(*c*) Exhaust occurs later; ceases earlier.
(*d*) Compression begins sooner; continues longer

(3) *Travel Increased.*

(*a*) Admission begins sooner; continues longer.
(*b*) Expansion begins later; ceases sooner.
(*c*) Exhaust commences ~~later~~; ceases later.
(*d*) Compression begins later; ends sooner.

(4) *Increased Regular Advance.*

(*a*) Admission begins earlier; period unaltered.
(*b*) Expansion begins sooner; period the same.
(*c*) Exhaust commences earlier; period unchanged.
(*d*) Compression begins earlier; period the same.

Thus: by adopting a lead of seven per cent of the valve-travel, a lap of twenty per cent in front and twenty-five per cent at the rear of the valve, the rod having a length of three times that of the stroke of piston, the normal admission is made seven tenths at both ends. The precise amount of lap or lead, or both, to be given in any case, can be determined by computation, but better by graphical methods or by model construction.

The effects of varying the lap, the lead, and the travel of the single three-ported slide-valve is well illustrated by the following table, in which a valve is taken as of $2\frac{1}{2}$ to $4\frac{1}{2}$ inches stroke, the lap from 0 to 2 inches, and the lead from 0 to $1\frac{1}{4}$ inches, for maximum stroke in the last two cases. The normal lap is 1 inch, and lead $\frac{5}{16}$.

Cut-off and compression are measured in fractions of full stroke. Variation of lap is evidently most generally satisfactory, but simultaneous variation of both lead and travel is the more usual method.

		Cut-off.	Compression.
Travel varying, lap and lead normal...	travel $4\frac{1}{2}$	0.74	0.09
	" $3\frac{3}{4}$	0.60	0.14
	" 3	0.40	0.25
	" $2\frac{1}{2}$	0.11	0.50
Travel and lead constant, lap variable.	lap 1 inch.	0.74	0.09
	" $\frac{1}{2}$ "	0.92	0.03
	" 0 "	1.00	0.00
Travel and lap constant, lead variable.	lead $\frac{5}{16}$	0.74	0.09
	" $\frac{5}{8}$	0.64	0.16
	" 1	0.50	0.30

Laying off the relative positions of crank, rods, and piston, as in the diagram below, it at once becomes evident that, as already remarked, in consequence of the angularity of the connecting-rod, the motion of the piston is not the same on both sides the middle of the cylinder. Assuming the crank to turn with perfect uniformity, it is seen that the motion of the pis-

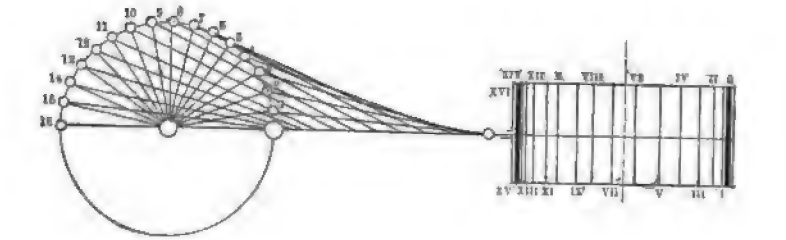

FIG. 90.—PISTON AND CRANK MOTIONS.

ton is most rapid in that part of its path furthest from the crank, and that it reaches the middle of the cylinder before the crank has traversed a quadrant of its path. In the first half-stroke the motion of the piston exceeds, in the second half is less than, its average speed. This irregularity is seen to be the greater as the rod is shorter; it can only disappear with a connecting-rod of infinite length.

The distance, d, of the piston from the middle of its path when the crank is at the half-centre, with varying proportions of rod to crank-length, $\frac{C}{R}$, is as follows, in per cent of stroke:

$\frac{C}{R}$	d	$\frac{C}{R}$	s
2	0.27	5	0.10
3	0.17	6	0.08
4	0.13	7	0.07

Studying the action of the valve with respect to one side of the piston:

Let r = the eccentric-radius;
R = the crank-radius;
l = outside lap of the valve;
l' = inside lap;
θ = angular advance of the eccentric.

Then the displacement of the valve from its middle position at the commencement of the stroke is seen to be

$$d = r \sin \theta;$$

the steam-port opening or steam lead is

$$o = r \sin \theta - l;$$

and the exhaust opening or lead is

$$o' = r \sin \theta - l' = o + l - l'.$$

When the crank turns through any arc ϕ, it carries with it the eccentric, and the total swing of the latter from the position corresponding to the middle of the valve-travel is $\theta + \phi$. Then the linear motion has become

$$d' = r \sin (\theta + \phi),$$

and the opening of the ports has been made, respectively,

$$o_1 = r \sin (\theta + \phi) - l;$$
$$o_1' = r \sin (\theta + \phi) - l' = o_1 + l - l'.$$

The corresponding movement of the piston, neglecting the angularity of the connecting-rod, is, S being the stroke,

$$s = R(1 - \cos \phi) = \frac{S}{2}(1 - \cos \phi).$$

Following the movement of the valve through the several phases of a complete cycle, we have:

(1) Crank on the centre; piston at the commencement of its stroke; then

$$\phi = 0; \quad d = r \sin \theta = l + o = l' + o';$$

and the steam- and exhaust-ports are opened, respectively, just the amount of the lineal steam and the exhaust lead.

(2) The crank has moved so far that the port is fully opened; then, if the widths of port are p and p',

$$p + l = r \sin (\theta + \phi);$$

and the piston, the obliquity of rod being neglected, has moved over

$$s = R(1 - \cos \phi).$$

(3) The eccentric-rod reaches the extremity of its throw. Then

$$d = r; \quad \phi = 90^\circ - \theta; \quad \theta + \phi = 90^\circ,$$

and the valve is about to retrace its path; while the piston has moved over

$$d = R[1 - \cos (90^\circ - \theta)].$$

(4) The return of the valve to the edge of the port repeats the conditions of the expressions in case (2), except that the piston has moved on.

(5) The valve reaches the external edge of the port, and, steam being thus cut off, expansion begins. Then

$$d = l;$$

and the exhaust remains open the distance

$$d' = l - l',$$

and the ratio of lap to eccentric-radius must be

$$\frac{l}{r} = \sin (\theta + \phi).$$

The piston has now moved over a space

$$s = \frac{S}{2} - R \cos \phi.$$

(6) The exhaust-port is closed and compression begins. Then

$$d = l'.$$

(7) The valve reaches its middle position, and is about to reverse its direction,

$$d = 0.$$

(8) The return of the valve removes the obstruction presented by the exhaust-lap on the opposite side of the piston, and

$$d = l';$$
$$- l' = r \sin (\phi + \theta);$$

the position of the valve being as in (6), and the piston very near the end of the forward stroke.

(9) The retreat of the valve commences opening the steam-port on the further side of the piston, and the engine takes steam with

$$d = - l.$$

(10) The valve regains its initial position, and the conditions of (1) are reproduced in every respect.

The crank-angles corresponding to various positions of the piston, where the connecting-rod is from 4 to 6 cranks' length, are given in the following table:

CRANK-ANGLES AND PISTON-POSITIONS.

Part of Stroke Completed.	Length of Rod in Terms of Crank.									
	4		4½		5		5½		6	
	Ahead.	Back.	Ahead.	Back.	Ahead.	Back.	Ahead.	Back.	Ahead.	Back.
0.1	33°	42°	34°	41°	34°	41°	34°	40°	34°	40°
0.2	48	60	49	59	49	58	49	58	50	57
0.3	60	74	61	73	62	72	62	72	62	71
0.4	72	86	72	85	73	84	74	84	74	83
0.5	83	97	84	96	84	96	85	95	85	95
0.6	94	108	95	108	96	107	96	107	97	106
0.7	107	120	107	119	108	119	109	118	109	118
0.8	120	132	121	132	122	131	122	131	123	130
0.9	138	147	139	147	139	146	140	146	140	146
1.0	180	180	180	180	180	180	180	180	180	180

The angle traversed by the crank between the points of admission and suppression or cut-off, or the equivalent angle between the points of exhaust-opening and compression, being given for any case, the ratios of lap to valve-travel, and of the

latter to the extent of opening of port, may be easily ascertained graphically, or, less easily, by computation. The following is a table of these quantities:

PROPORTIONS OF SLIDE-VALVE.

Angle of Movement.	Ratio of Lap to Travel.	Ratio of Travel to Port-opening.
45°	0.46	26
50°	0.45	21
55°	0.44	18
60°	0.43	15
70°	0.41	11
80°	0.38	8.6
90°	0.35	6.8
100°	0.32	5.6
110°	0.29	4.7
120°	0.25	4.0
130°	0.21	3.5
140°	0.17	3.0
150°	0.13	2.7
160°	0.09	2.4
170°	0.04	2.2
180°	0.00	2.0

Setting the Slide-valve is a simple process, but one which demands some care; in this operation the engine is first set exactly "on the centre," and the valve is set and rods adjusted to give precisely the desired steam-distribution.

In setting the engine on the centre, the brasses are all set up so that no looseness or play in bearings can affect the several motions. A line is "scribed" on the guides and crosshead in such manner as to permit of their being restored, at pleasure, *precisely* to the same relative positions, and with the crank 30° or 45° above or below the centre. The arc traversed by the shaft, crank, or wheel-rim, in turning until the coincidence of these marks is restored being noted, and the half-arc then being moved over, the engine is "on the centre." Marks

on the wheel, or on the crank-disk, guide in the measurement of this arc and in its bisection.

With the crank on the centre, the eccentric is set in position, the valve secured on its stem and with the desired lead. Turning the engine ahead, the lead on the opposite centre is measured, and, if not exact, the valve readjusted on its stem to make the lead equal at both ends, and the eccentric is then moved to make its amount right.

The eccentric is given usually a throw such that the travel of the valve will be at least sufficient to uncover the port fully at each movement; the eccentricity, with direct connection, being thus equal to the sum of the lap and the port-opening. The outside or steam lap is variable between wide limits, but is usually not less than one fourth nor more than two thirds the proposed port-opening; a common proportion being one third the latter dimension, and, therefore, one eighth the total valve-travel. The inside or exhaust lap is very generally made not over one tenth the port-opening; and many designers use none, or even give "negative lap." The precise amount is best determined by laying down the desired form of indicator-diagram, and constructing a Zeuner diagram from it, thus securing a measure of all the characteristics of the valve-motion.

Common values of the linear steam-lead are thus about one eighth to one tenth the port-opening; and since the relation holds

$$\sin \theta = \frac{l + l'}{r},$$

it is easy to determine the angle of lead. Thus, when $l = \frac{1}{8}$ and $l' = \frac{1}{5}$, the radius of eccentric being 1,

$$\sin \theta = 0.3 + ; \quad \theta = 19^\circ + .$$

The angular advance of the eccentric is always $\theta + 90^\circ$.

The introduction of a rock-shaft reverses the relative motion of valve and eccentric, compelling the alteration of the position of the latter to a point diametrically opposite that other-

wise given it; in such manner that it may, by this reversal of movement, give the right motion to the valve.

It is obvious that increasing the travel of the valve by increasing the throw of the eccentric will permit either increased lap or a quicker cut-off action. Removing lap will effect the latter, the eccentricity being retained unchanged.

The two usual methods of adjustment of the steam-distribution of the single valve have been seen to be: (1) shifting the eccentric on the shaft; (2) swinging it across the shaft. In the first case, the result is simply altering the lead, or the time of every event, the throw of the eccentric and valve remaining unchanged; in the other case, the travel of the valve, and often the lead as well, are altered, and the distribution modified by the change thus produced in the proportions of the motion to those of the valve and its lap.

In the common locomotive or Stephenson gear, the motion is shifted from one eccentric to another by means of the link, the result being similar to that due to the shifting of an eccentric from the one position to the other; except that, at intermediate points, the throw is reduced. In the Armington and Sims gear, the same effect is obtained by a pair of symmetrically placed eccentrics which are together shifted toward or away from each other by the governor. In many other engines the eccentric-centre is swung across the shaft-centre by making it a part of an arm pivoted at one side the shaft in such manner that the path of the eccentric is an arc lying across the shaft in the desired position. In other designs the eccentric is loose on the shaft, and shifted so as to vary its lead by the action of the regulating mechanism. Nearly all the "high-speed" engines in the market, and especially designed for driving electric-lighting machinery, illustrate one or another of these types. In the old "Dodd motion," and in one or two later designs, the eccentric is thrown across the shaft in a straight line by the use of wedges working in a slot in the eccentric, in such a manner as to give a variable throw.

With the Stephenson gear, the twin shifting eccentrics, and the Dodd motion and their equivalents, the eccentric-centre,

real or virtual, being shifted in straight lines across the shaft, the lead may be kept constant, while the other elements of the distribution change. The same result may be approximated with the swinging arm; but in any case it is usually better, probably, to give less lead at high ratios of expansion when the compression is large. In the practice of some engineers it is even found desirable here to make the lead negative.

Where a shifting-link is employed, and consequent variability of cut-off is secured, the travel of the valve is made greater, as the higher ratios of expansion are expected to be more generally used, in order that a full and free opening may be then secured.

The following are the usual simple designer's rules and tabulated quantities as given by Molesworth:

$W =$ width of the steam-port in inches;
$l =$ lap in inches;
$L =$ lead in inches;
$S =$ stroke of engine;
$T =$ travel of the slide;
$X =$ the distance travelled by the piston before the steam is cut off;
$y =$ distance of piston from end of stroke before exhaust is cut off;
$z =$ distance of piston from end of stroke before exhaust begins;
$l' =$ eduction overlap;
$t =$ linear advance $= L + l$;
$\theta =$ angle of advance;
$\theta' =$ angle of eccentric;
$\phi =$ angle of crank $= a + 90° - \theta'$;
$\phi' =$ angle of eduction overlap.

$$\sin \theta = \frac{2t}{T}; \quad \cos \theta' = \frac{2l}{T}; \quad \cos \phi' = \frac{2l'}{T};$$

$$X = \frac{S}{2}(1 + \cos \phi);$$

$$y = \frac{S}{2}[1 - \cos(\theta + 90^\circ - \phi')];$$

$$z = \frac{S}{2}[1 - \cos(\theta + \phi' - 90^\circ)];$$

$$X = S\left[1.0 - \left(\frac{2l + L}{T}\right)^2\right];$$

$$l = \left(\tfrac{1}{2}T\sqrt{\frac{S - X}{S}}\right) - \tfrac{1}{2}L;$$

$$T = 2l + 2W.$$

LAP REQUIRED AT DIFFERENT CUT-OFF PARTS OF THE STROKE.

Travel of Slide.	Portion of Stroke in Full Stram.								
	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{5}{6}$	$\frac{7}{8}$
ins.									
4	1.73	1.63	1.58	1.41	1.22	1.15	1.00	.82	.71
6	2.60	2.45	2.37	2.12	1.84	1.73	1.50	1.22	1.06
8	3.46	3.26	3.16	2.83	2.45	2.31	2.00	1.63	1.41
10	4.33	4.08	3.95	3.53	3.06	2.89	2.50	2.04	1.77
12	5.20	4.90	4.74	4.24	3.67	3.46	3.00	2.45	2.12
14	6.06	5.71	5.53	4.95	4.28	4.04	3.50	2.86	2.47
16	6.93	6.53	6.32	5.65	4.90	4.62	4.00	3.26	2.83
18	7.80	7.35	7.11	6.36	5.51	5.19	4.50	3.67	3.18
20	8.66	8.16	7.90	7.07	6.12	5.77	5.00	4.08	3.54
22	9.53	8.98	8.69	7.78	6.73	6.35	5.50	4.49	3.89
24	10.39	9.80	9.48	8.48	7.35	6.93	6.00	4.90	4.24

NOTE.—If the lead is given, one half its value must be deducted from the numbers above tabulated.

The angular vibration of the connecting-rod, causing inequality of motion of the piston on either side the points at which the crank and rod are at right angles, introduces a resultant unsymmetrical distribution of steam, the cut-off, in direct-acting engines, being thus deferred on the forward-stroke and shortened on the return-stroke; every event in the cycle of the engine being thus made to occur in changed time, as the length of the rod is reduced from infinity, except the passage of the centres and effects dependent thereupon. The

rod thus brings the piston nearer the crank, as it is shortened, at all parts of the stroke between the centres. The uniform speed of rotation of the crank thus, also, makes the speed of piston greater on the forward-stroke than on its return.

These distortions are sufficient to make it necessary to provide against them in designing the valve-gear.

Similar distortions, due to drawing the valve toward the shaft, are produced by the vibration of the eccentric-rod; but they are usually comparatively unimportant, the ratio of length of rod to its travel being large. The eccentric-rod is slightly lengthened to correct the evil, and to retain nearly equal lead at both ends. Adding exhaust-lap on the end of the valve toward the shaft, and introducing a similar "negative lap" on the outer end, corrects the action of the exhaust side. This cannot, however, be done on the steam side without producing an altered and unequal lead. This latter is usually considered a serious fault, although it is less important as the lead is greater. By the introduction of an intermediate rock-shaft, with its driving and driven arms somewhat out of line, and with its pins off the direct line between shaft and valve, a correction may be effected free from this objection.* Compression and exhaust may be similarly equalized.

With varying expansion, a shifting-link or equivalent mechanism being used, the lead of the valve is commonly also variable and may become unequal at the opposite ends. The latter is probably, in most cases, the more objectionable of these two effects. Where the eccentric is movable to secure varying angular advance, or to obtain a variable throw, the rod is also caused to produce a variable effect by its altered angularity; which is less with larger expansion, and greater with a low value of that ratio. These conditions may readily be traced out by graphical construction, and are best studied on the drawing-board in skeleton representations of the valve-motion.

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* See Am. Machinist, March 14, 1889; also Halsey's Slide-valve Gears, p. 54.

The equalization of the cut-off is effected by Professor Sweet by the same device as that which permits equal lead,—the employment of an unsymmetrically-set rock-shaft. That gear is given an increasing lead with increasing expansion-ratios, in order to secure a practically uniform compression with variable expansion.*

54. **Graphical and Geometrical Solutions of Slide-valve Problems** are much more satisfactory than algebraic, as the latter are too complicated and cumbersome for convenient application in practice. The following are simple methods of treatment:

When the obliquity of the connecting-rod, as well as that of the eccentric-rod, may be neglected, we may adopt the following methods to find the proper lead and lap of the valve:

First method.—About a centre O describe the circle $DEFI$, and draw $\overline{DF}$ and $\overline{EI}$. Consider $\overline{DF}$ as the stroke of the piston; and $\overline{EI}$, on a different scale, that of the valve; and let motion from D to F be considered as a forward-stroke.

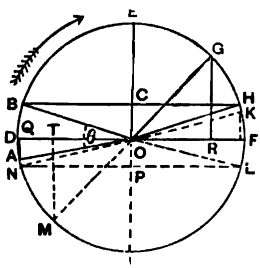

FIG. 91.

Let Q represent the point of the return-stroke where the admission is to begin. If the admission and the forward-stroke are to begin together, Q will coincide with D.

Let R be the point where the steam is to be cut off.

Let T be the point of the return-stroke where compression or cushioning is to begin.

(1) *To find the angle of lead and the steam-lap.* Draw QA, RG, perpendicular to DF; measure or bisect the arc AG; from E lay off the arcs EB, EH, each equal to one half AG; join BH parallel to DF.

$$\text{The angle of lead} = \angle AOB = \angle GOH; \quad . \quad . \quad (3)$$

$$\frac{\text{Lap on the steam side}}{\text{Half-throw}} = \frac{\overline{OC}}{\overline{OE}}. \quad . \quad . \quad . \quad . \quad (4)$$

* For a concise description of such methods, see Halsey's Slide-valve Gears.

(2) *To find the lap on the exhaust side and the point of release.* Draw TM perpendicular to DF, cutting the circle in M, and lay off the arc $MN = AB$. Draw NL parallel to DF, cutting OI in P; then

$$\frac{\text{Lap on the exhaust side}}{\text{Half-throw}} = \frac{\overline{OP}}{\overline{OE}}. \quad . \quad . \quad . \quad . \quad (5)$$

from L lay off the arc $LK = AB$, and from K draw KS perpendicular to DF; S represents the *point of release* during the forward-stroke, or that at which the valve begins to open on the exhaust side.

Second method.—By trigonometrical calculation.

Data:

$$\frac{\text{Advance of admission}}{\text{Stroke of piston}} = \frac{\overline{DQ}}{\overline{DF}} = \frac{1}{q};$$

$$\text{Ratio of actual cut-off} = \frac{\overline{DR}}{\overline{DF}} = \frac{1}{r'};$$

$$\text{Ratio of cushioning} = \frac{\overline{DT}}{\overline{DF}} = \frac{1}{r''};$$

Half-throw of slide-valve, OE.

Results:

Let angle of lead $= a$;

angle of lap on induction side $= b'$;

angle of lap on eduction side $= b''$.

Then

$$a - b' = \cos^{-1}\left(1 - \frac{2}{q}\right); \qquad a + b' = \cos^{-1}\left(\frac{2}{r'} - 1\right);$$

$$a + b'' = \cos^{-1}\left(1 - \frac{2}{r''}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Then we have

$$a = \frac{(a + b') + (a - b')}{2}; \quad b' = \frac{(a + b') - (a - b')}{2};$$

$$b'' = (a + b'') - a; \quad \ldots \ldots \ldots \ldots \quad (7)$$

and also,

$$\left.\begin{array}{l} \text{lap on induction side, } \overline{OC} = \overline{OE} \cdot \sin b'; \\ \text{lap on eduction side, } \overline{OP} = \overline{OE} \cdot \sin b''. \end{array}\right\} \quad . \quad . \quad (8)$$

Fraction of stroke at which release occurs,

$$\frac{\overline{DS}}{\overline{DF}} = \frac{1 + \cos (a - b'')}{2} \quad \ldots \ldots \quad (9)$$

To take into account the obliquity of the rods, use one of the foregoing methods to find the angle of lead. Then make a drawing, on a large scale, showing the crank in a series of equidistant positions. The lead being known, the positions of the eccentric-radius may be laid down. Draw the centre lines of the piston and valve-rod, and lay down the positions of the piston and the valve corresponding to the given positions of crank and eccentric. The number of positions employed may be from twelve to twenty-four; they are numbered in their order.

Then draw to the same scale a diagram, in the following

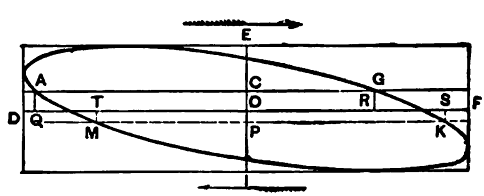

FIG. 92.—VALVE-DIAGRAM.

manner: Draw the axes DF, EI, bisecting in O. Make $\overline{OD} = \overline{OF} =$ the half-stroke of piston, and $\overline{OE} = \overline{OI} =$ the half-throw of the valve. On DF mark points corresponding to

the positions of the piston found on the drawing; and lay off ordinates parallel to EI, $\overline{TM}$, etc., representing the corresponding distances of the valve from its middle position. Through the ends of these ordinates draw the oval curve $MAGK$. This oval figure corresponds to the forward-stroke in its upper, and to the backward-stroke in its lower, half.

Then mark the points of cut-off R, and of cushioning T; draw ordinates RG, TM, perpendicular to DF, cutting the curve in G, M. Then

$$\text{the lap on the steam side} \quad = \overline{RG};$$

$$\text{the lap on the exhaust side} = \overline{TM}.$$

Draw GA and MK parallel to DF, cutting the curve in A and K, from which points let fall on DF the perpendiculars AQ, KS. Then will Q be the point of the stroke at which the admission begins, and S the point of release.*

Sometimes, in vertical engines, the lap of the valve on the steam side is made less for the lower than for the upper end, to cut off steam later, and to secure a less ratio of expansion and greater mean pressure during the up-stroke than during the down-stroke, in order to equalize the work done on the crank during the two strokes. The difference of total pressures should, in such cases, be equal to twice the weight of the piston, piston-rod, and connecting-rod, which weight assists the down-stroke and resists the up-stroke.

The Zeuner Diagram, as devised by its author, for the purpose of solving problems relating to steam-distribution by means of the slide-valve,† is the most common, as it is one of the best and simplest of graphical construction of this class. By this device it is easy to determine the relative motion of valve and crank, and then, knowing the length of connecting-rod, to determine the motion of the piston relatively to the

* Numerous examples are given in Mr. D. K. Clark's work on Railway Machinery. See also Rankine, p. 493.

† "Die Schiebersteuerungen": G. Zeuner, 1868. See also "Designing Valve-gears." E. J. C. Welch, London, Spon, 1875; and "Die Steuerungen der Dampfmaschinen," von E. Blaha, Berlin, 1885.

valve. By the use of this diagram, the position of the eccentric relatively to the crank being known, and the lap and lead given, the points of admission, of cut-off, of exhaust, and of compression become readily determinable in their relation to the arc traversed by the crank; and the converse set of problems can be solved; while, in each actual case, the position of the crank fixes the corresponding position of the piston at the several important periods in steam-distribution.

Considering the movement of the crank and eccentric, it will be seen that the arc moved over by both, from the instant of opening to that of closing the steam-port, is the same, both being usually keyed to the same shaft, or, rarely, to separate shafts moving synchronously. The arc traversed by the crank during this period being the same as that described by the eccentric, and being determined by the times of admission and of cut-off, it becomes easy, knowing the lead and the point of cut-off, or ratio of expansion, to ascertain the corresponding motion and the required lap of the valve. When without lap or lead, the eccentric is set at right angles with the crank and describes a semicircle during the period of admission of steam. When steam is cut off at any given point, the valve must open and close while the eccentric traverses the arc measured off by the crank between the instant of admission and of cut-off; and one half this arc measures the swing of the eccentric while opening the port, as also while closing it; evidently, the greater the expansion the less this arc; the more the lead given, the greater the arc. Thus, in the figure, if the engine takes steam at A and "follows full-stroke," the crank traverses the arc ADG, and the eccentric moves through the arc AD while opening and through DG while closing the port. If steam is cut off at E, the arc ADE is the measure both of the crank and the eccentric-movement, and the latter moves over an arc equal to, but not coincident with, AC while opening, and

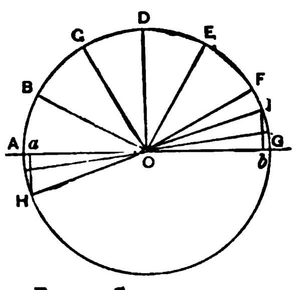

FIG. 93.—CRANK-MOVEMENT.

CE while closing it. Were it practicable to cut off at C, the valve would be opened while the eccentric traversed the arc AB and closed as it moved through BC. Lead being given, as at H, the arc HA is added to the arcs of admission, and their totals become HDI, HCE, and HBC. Since the valve cannot open during more than a quarter-revolution of the shaft, the lead at H produces, for the case of full-throw, a corresponding negative lead at closing, at IG, the magnitude of the lead being Aa or bG.

In the next figure, let the first arrangement be represented. The valve is moved over a distance OD by the eccentric, as before, while the latter traverses the arc ADG. In this case the port is opened the instant the eccentric moves from A toward D, the start being made from its middle position, the valve-travel being KD, and the eccentric being set at right angles with the crank. When steam is cut off at E, and the angle described is reduced to ADE, the valve must begin to open at distance, BD, from D, equal to one half the full arc, and must close while traversing the equal distance, DF; but since it moves over AB and FG, retaining a closed port, it is evident that "cover" or "lap" must be given by building out the edge of the valve to the extent measured by $r \sin AB$, or by OH, as already seen in Fig. 91, above. The half-travel of the valve is thus equal to the sum of the lap, OH, and the port-opening usually, but not always, the width of the steam-port, HD. The half-difference between the arc traversed with open port and the semi-circumference thus measures the angle of lead of the eccentric, i.e.,

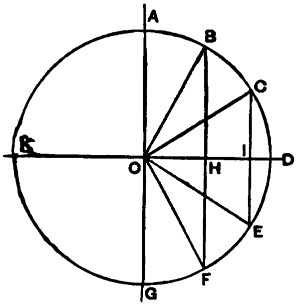

FIG. 94.—VALVE-MOVEMENT.

$$AOB = ADG - BDF.$$

Where lead is added, the angle of lead is increased by the amount of the angle traversed to give the required lead, and

the arc BDF must be that moved over by the crank from the instant of "preadmission" to the point of cut-off.

In Zeuner's diagram, the crank- and valve-motions are both so represented that it is easy to solve all problems arising in the designing of this form of valve-gear.

Referring to the last figure, it is seen that we have

$$\frac{\text{half-travel of valve}}{\text{lap}} = \sin AOB = \frac{OB}{OH}.$$

The angle OHB is a right angle, and a semicircle constructed on OB as a diameter will pass through H, measuring off the lap OH. Setting the eccentric ahead to secure lap and lead throws the position of crank, at mid-stroke of the valve, backward an equal amount. In the last figure, setting the eccentric ahead by the angle AOB reduces the arc traversed, while the port is open to steam, to BDF, equal to the crank-movement ACE in the preceding figure, the middle of which latter arc is on the radius OD, and the angle COD of the last is equal to AOB on the first of the two diagrams; their sines are equal, and the same construction may be adopted on both radii to obtain the ratio of lap to half-travel, and to solve related problems.

Since the distance travelled by the valve from its middle position is always measured by the base Oe, Of, Og, Fig. 95, of the triangle described within Zeuner's auxiliary circle, the progress of the valve, when set at right angles to the crank, as here, can always be traced by drawing the successive crank-positions, OA, Oa, Ob, Oc, etc., and measuring the intercepts, Oe, Of, Og, etc.

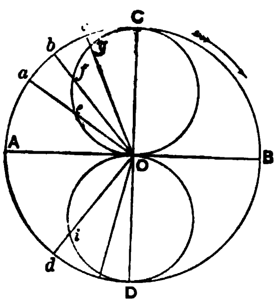

FIG. 95.—VALVE-TRAVEL.

Then, since these last quantities measure the distance so travelled by the valve, they also measure the successive magnitudes of port-openings as the crank swings through these arcs. It is further evident that if the valve have lap equal to Oe, the valve would just begin to

open at the crank-position *Oa*, and similarly of *Of* and *Ob*; so that the line *Oe*, *Of*, or other, always measures the lap needed to secure a deferred opening; or, what is the same in effect, the lap needed to produce cut-off at the same angular distance from the opposite "centre," the eccentric being then set ahead in such manner as to begin to uncover the port at the beginning of the stroke.

The difference between *Oe*, or *Of*, or *Og*, etc., and the full travel *OC*, measures the maximum opening in the latter cases, lap being used. As the valve is here set, the exhaust-opening can be similarly traced, and the port is open to the full extent, *OD*, when the crank is vertical.

The distance traversed by the piston during any given angular movement of the crank is readily determined by the graphical construction illustrated in the accompanying figure.

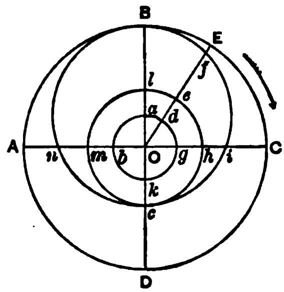

FIG 96.—PISTON-MOVEMENT.

Let *O* represent the position of the crank-shaft centre; let *Oa* be the length of the crank, and let *aB* be the length of the connecting-rod. Then the line of the engine being assumed to be *OB*, and the piston at the far end of the cylinder, any point in the circle *ABCD* will be at a distance from the shaft-centre equal to the sum of the lengths of crank and rod. Now, suppose the whole system to be swung about *O*, except that the crank remains at rest at *Oa*. When a half-circle has been described, the piston, still held at the same distance from the crank-pin, will now be nearer *O* by the distance *aOk*, the length of stroke, and will have moved toward *O* this distance, equal to *Dc*. The two circles *ABCD* and *lhcm* are thus always separated by the length of stroke. But the circle *Bicn* represents the orbit of the crosshead-pin, and its distance, radially, from the external circle must always thus measure the path moved over, up to the given or assumed radius, by the piston, the latter having started at *B*. Thus, draw the radius *OE*; then *eE* is the troke of piston; *fE* is the distance

traversed by it while the arc *BOE* is described, and *ef* is the distance to be completed before reaching the end of the stroke.

But the same quantities are measured off if it is assumed that the crank moves and that the arc *BOE* is described by it; and each radius drawn in its position thus identifies the distance traversed by the piston, as *Ci* at the half-centre, *Dc* at the opposite centre, and *An* at three quarters of a revolution.

The relation of the Zeuner diagram to the older graphical construction is such as to afford a means of presenting a simple and exact proof of the accuracy of the former.

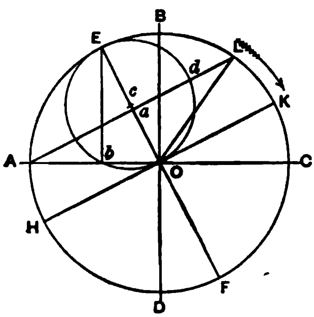

FIG. 97.—THE VALVE-DIAGRAM.

In the figure, let *ABCD* be the crank-circle; let *AC* be the centre-line of the engine; let the steam-port open at *A* and close at *L*; and, consequently, let *EO* be the position of the crank when the valve is at its end-position and about to reverse its direction of motion. Then, since the valve must begin to open the port at *A* and must close it at *L*, the extent of opening is measured by *Ea*, and the lap, or cover, by *aO*. The valve, starting from its mid-position at *O*, moves out the distance *Oa*, the lap keeping the port closed until, at *a*, the edge of the lap begins to uncover it; and, from *a* outward, the travel to any given point, as *c*, less the lap, measures the port-opening; provided, as should always be the fact, the width of port in the line of motion of the valve is not less than *aE*.

Next describing the circle *ObEd*, the Zeuner figure is produced, and *Ob* is the needed lap. For, erecting *Eb* at *b*, we have the two right-angled triangles *OaL* and *ObE* similar, the sides *OL* and *OE* and the angles *aOL* and *aOb* being equal; and hence the triangles are equal, and, *Oa* being equal to *Ob*, the latter, like the former magnitude, is the measure of the lap demanded to secure the required expansion and cut-off.

It can in the same way be shown that the Zeuner diagram may be made to exhibit the action of the exhaust side.

(1) *To find the lap required, the proposed distribution being given.* In the next figure let the position of the crank when the engine takes steam be OH, and that at cut-off be OG; then, bisecting this angle, OE is the position of the crank when in the middle of the range of both crank and eccentric motion during the admission period. The angle COE measures the recession of the crank, with increase of lap, and the advance of the eccentric, with variation of this mid-swing of both, during that period.

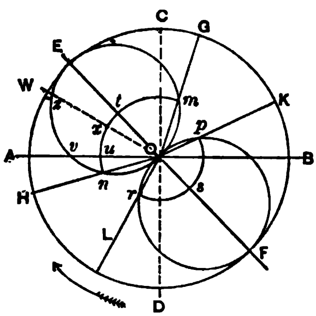

FIG. 98.—THE ZEUNER DIAGRAM.

With the radius Om describe the arc mtn, and the ratio $\frac{Om}{OE} = \frac{On}{OE}$ is that of the lap required to the half-travel of the valve.

(2) *To find the linear lead.* On the diagram, as drawn, the distance Ou is the lap of the valve required to give steam at the crank position OH and to close the port at OG; but it will be seen that Ov is the lap needed to open the port at the beginning of stroke, the crank at OA; then uv must be the opening of the port with the given lap Ou, i.e., the linear measure of the lead on the assumed scale. The position of the edge of the valve, relatively to that of the port, is always measured by the breadth of the lune $nvEmtu$ taken on the radius representing the corresponding position of the crank, and varies from 0 at n to its maximum, tE, at the middle of the arc HEG. Thus, when the crank is at OW the port-opening is xz.

(3) *To find the exhaust-lap and beginning of compression.* Extend EO to F, making OF equal OE, and describe the circle $OpFr$, making its diameter equal to the half-travel of the valve, as before. Let the position of the crank when the exhaust opens be given, as OK; then $Op = Or$ will measure the lap on

the eduction side of the valve, and $\frac{Op}{OF}$ its ratio to the half-travel.

The opening being fixed, the closure must evidently take place when the crank has swung through the arc $HEG = KFL$; and the arc *psr*, described with the radius *Op*, locates a point *r*, and the radius *OL*, and defines the position of the crank when compression, or cushioning, begins.

The reverse construction gives the location of *OK* and the point of exhaust-opening when the datum given is the point at which compression begins, and also the lap needed to secure the desired action of the valve.

(4) *The distribution of steam is to be found, the valve being given.* Draw *AB* and *CD* at right angles; draw *EF*, making *EOC* equal to the angular advance of the eccentric; draw the circles *OmEn*, *OpFr*, as before; take *Ot*, *Os*, as radii measuring the given lap of valve on steam and on exhaust side, and describe the arcs *mtn* and *rsp*; finally, draw the radial lines *OA*, *OG*, *OK*, and *OL*, and they will respectively represent the positions of the crank at admission, at cut-off, at exhaust, and at the beginning of compression.

When the steam-distribution is intended to be different at the two ends, the construction is made for each separately. The position of the crank being determined, as above, for each point of opening or of closing, the position of the piston is easily obtained by laying down the movement of crank, connecting-rod, and piston on the drawing-board to a convenient scale.

The several phases of action of the plain slide-valve, as described in § 53, are also exhibited on the Zeuner diagram, thus:

In the figure let *AB* be the centre-line of the engine and that of motion of the valve, and let *CD* be drawn perpendicular to it. Make *COE* the angle of advance of the eccentric and *OR* its radius. Draw the valve-circles *OVRU* and *OSFT*. Then *ON* is the steam-lap and *OW* the exhaust-lap, while *NX* is the measure of full port-opening; the sum of *ON* and *NX*

being the extreme movement of the valve from its middle position.

(1) The first phase is that in which the piston is on the

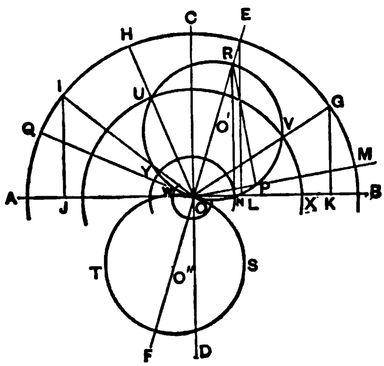

FIG. 99.—ACTION OF THE SLIDE-VALVE.

centre and the eccentric at the angle *COE* ahead of the half-throw position, the lead on the steam side being *NL*.

(2) The second phase is that in which the crank and eccentric have moved through an angle *BOG*, moving the valve a distance, *OV*, nearly equal to the eccentric throw, and completely opening the port. The valve is now nearly on the point of starting on its return.

(3) The third place is indicated by the angle of crank-movement *BOE*; the extreme throw of the valve is reached, and it is ready to return.

(4) The fourth step takes the crank over to *OH*; the port is on the point of becoming again partly covered; $OU = OV$.

(5) Fifthly: the crank swinging over to *OI*; *OV* measures the distance of the valve from its middle position and also the steam-lap, and the port is just closed, expansion beginning at this phase.

(6) The crank passes over to *OQ* and the exhaust-lap *OW* is sufficient just to cover the exhaust-port, and compression begins.

(7) When the crank-movement has become $\theta + \phi$, and its

line is a right angle to OE, the valve has again reached its middle position.

(8) When, the displacement of the valve continuing, the effect of the exhaust-lap is wholly removed from the steam side, the port opens, and the steam is exhausted from the cylinder.

(9) Finally, the crank having passed over the arc 180°, less the lead-angle, steam is taken on the opposite side of the piston.

(10) The crank having assumed the position AO, a similar cycle commences for the side of the piston opposite that now under consideration.

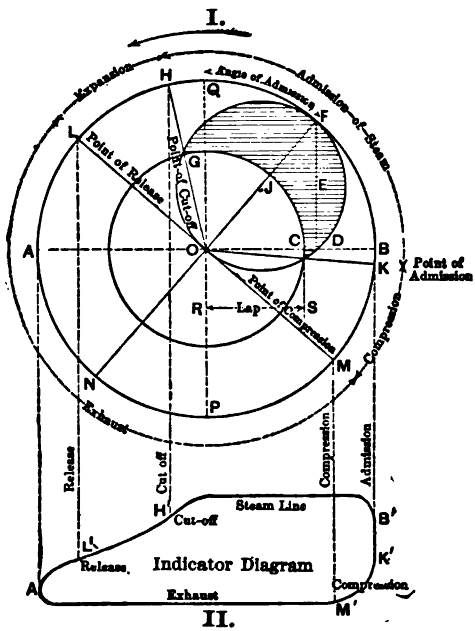

FIG. 100.—STEAM-DISTRIBUTION.

Fig. 100 exhibits the method of ordinary steam-distribution still more clearly by its comparison of the valve-diagram I with the indicator-diagram II ; and the several corresponding phases in the engine and valve cycles are designated

on both these diagrams. In diagram I the line of piston-travel is AB; that of valve-movement, OF, being set back by the angle of admission, QOF, from the vertical; while we have

$$JOK = JOQ,$$

the limit of travel occurring at OF. CD is the steam-lead; OD the total movement from zero when the centre is on the centre; JF is the width of the port, which is fully open when F is reached. OGH is the position of the crank when the port again closes and cut-off takes place; OK and OH being placed symmetrically with respect to OF. L and M are the points of release and compression, the line LM being necessarily at right angles to NOF, and marking the position of the crank when the valve-edge is at the edge of the port, there being here no internal lap. The external lap is OJ or RS, and the valve must move so far from its middle position to take off the lap and to begin uncovering the port.

Diagram II shows each of the critical points in the cycle, as revealed, less precisely, by the indicator, and the two may be compared with profit. Given I, we may construct II; given II, we may often obtain a fairly approximate construction of I.

In designing the slide-valve, in detail, the process is substantially as follows:

Let the dimensions of the engine be

Diameter of steam-cylinder...............	20 in.
Stroke of piston........................	36 "
Length of connecting-rod................	8 ft.
Width of steam-port.....................	2 in.
Maximum cut-off.........................	30 "
Admission before end of stroke..........	½ "

We are to determine the lap and lead and position of the eccentric relatively to the crank.

In Fig. 101, let O represent the centre of the main shaft; make OX the centre line of the engine, and draw the perpendicular OY. Describe the circle $ABCD$ to represent the path

of the crank-pin, and make *CX* the length of the connecting-rod, and *XS* the stroke of piston. Measure off from *X* to *E* the point at which steam is to be preadmitted, and draw *EL* equal to the length of rod, to determine the point, *L*, in the motion of the crank at which admission takes place, and the valve begins to uncover its port. Next, measure from *X* to *G* the distance to the latest point of cut-off, 30 inches, and similarly locate *H*, the position of the crank at that instant, and thus obtain a measure of the angle, *COH*, through which the crank swings while the valve is opening and again closing the steam-port. Bisecting this angle by the radial line *OI*, the angular advance of the eccentric is found to be *COI*, and it is set ahead of the normal position where neither lap nor lead is given, at

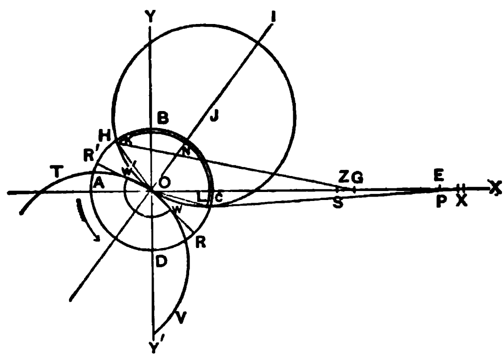

FIG. 101.—DESIGNING SLIDE-VALVES.

right angles to the crank, by that amount. From *J*, a point on *OI*, as a centre, describe the valve-circle, *FIH*, cutting *OH* and *OF* at points equidistant from *O* in *K* and *L*; and the arc *LNK* drawn through this point will be the lap-circle, and will measure the lap of valve needed to effect the cut-off as proposed, *ON* being the lap and *NI* the width of port; or, should the scale be altered, these two distances would measure the proportion of lap to the half-travel of the valve.

Similarly, for the exhaust side, if *P* be the point in the stroke at which compression begins, the corresponding point in the path of the crank, *R*, may be identified and, the other valve-circle *TOV* being drawn, cutting *OR*, *OR'*, in *ww'*, gives

the exhaust-lap as Ow; while the length of rod measured from R' locates Z as the point in the stroke at which exhaust begins.

The experimental use of Zeuner's or other construction in this manner will illustrate well the fact that the single valve cannot be used to give a high ratio of expansion; since it gives too early exhaust and too great compression.

A careful examination of these graphical methods will show plainly that they are applicable to nearly all practical arrangements of valves and their gearing when driven by eccentrics; and their use is so convenient, and they are so fruitful both in the solution of defined problems and in suggestions of real value, that the designing engineer will usually make their application a matter of extended and deliberate study.*

Uhland has devised a very convenient method of exhibiting the movement of the slide-valve, which is illustrated in the figure, the outline of the valve being shown in its middle position and the parallel lines below it indicating its successive displacements as the piston moves forward and backward. The stroke is divided into tenths, and each line shows the location of the valve-face at the corresponding point in the stroke. L and L' show the lead; and the breadth of opening of port at each step is readily measured.

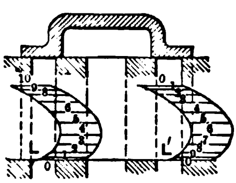

FIG. 102.—UHLAND'S DIAGRAM.

A comparison of the three principal forms of graphical representation of the motion of the slide-valve and its effects will serve to clearly exhibit the characteristic features of each. In the following illustrations are shown the Zeuner diagram (A), the standard form of ellipsoidal curves (B), and the continuous sinusoidal curves obtained by assuming a point to be carried in one line by the motion of the valve, and in a line at right angles with the former by the movement of the piston (C).†

* See Designing Valve-gearing, E. J. C. Welch, 1875; Slide-valve Gears, H. Bilgram, 1878; Slide-valve Gears, F. H. Halsey, 1890.

† See Busley: Die Schiffsmaschine; Atlas.

In the first of these various representations of the same events and same construction, *ab* is stroke of the engine, *cOd* the vertical line in which the crank stands at half-stroke, *ef* the line of extreme throw of eccentric when the crank is on the centre and when *ab* is the centre-line of the engine. *Oe* is the radius of eccentric-throw, as *OA* is that of the crank; *gf* is the

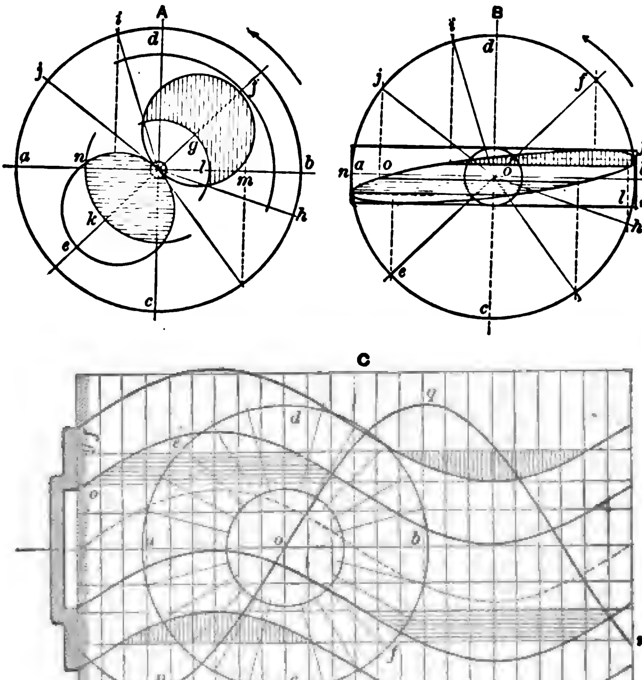

FIG. 103.—COMPARISON OF DIAGRAMS.

steam, and *ek* the exhaust, lap. Steam enters when the crank is at *Oh*, is cut off at *Oi*, and exhaust occurs at *Oj*. The steam-lead is measured by *lm*, the exhaust-lead by *nO*.

In diagram *B*, the same events and measures are indicated by the same symbols, as also in *C*. In the latter, *pqr* is the sinusoid described by the piston as it moves in its path, the paper travelling under it uniformly in a direction at right angles

to the movement of the cross-head. The other sinusoids illustrate the motions of the several parts of the valve, at which these lines terminate, during the synchronous motion of the piston. The shaded parts exhibit the port-openings. The rectilinear diagram and the circle are divided into similar and corresponding equal parts; and, the two being compared, it is easy to follow the motion, to compare this with the preceding diagrams, and to perceive the significance of every line in either.

55. **Expansion-valves** often give a much better distribution than the single valve. In designing the single three-ported valve, the engineer is embarrassed by the interdependence of the several events of the steam-cycle. A shorter cut-off means a great ercompression; increasing lead affects every other element of distribution; early or late exhaust makes all other events, other proportions of the valve being unaltered, occur comparatively early or late; and the design is always the resultant of a compromise of conflicting requirements. It is not usually practicable to arrange a scheme of distribution and construct the valve-motion to precisely produce it.

The main embarrassment is, however, the necessity of securing considerable expansion; which compels submission to the disadvantages, often, of excessive compression, and of long stroke of valve and corresponding waste by friction. The exhaust side is always controlled by the steam side. By the use of a separate, an independent, expansion-valve, the designer can adjust it to give any proposed ratio of expansion, and is then left free to proportion the main valve to any distribution in other respects that he may consider necessary or desirable, and the exhaust side can be made to suit his requirements in giving free discharge and the desired compression. The release is commonly a few degrees in advance of the end of stroke, but is greater in fast engines, sometimes being as early as 0.9 stroke, and the steam-lap is thus made small, necessarily, and without detriment, the cut-off by the lap on the main valve occurring late, and after the action of the cut-off valve.

The cut-off valve is sometimes, as in Meyer's gear, a pair of blocks, sliding on the back of the main valve, and separated or

approximated by a hand-adjustment to cause a more or less early closing of the ports cut through the latter, over which they slide. This is a case of variable lap and fixed travel. When the blocks are set close together, their stroke is insufficient to carry either over the adjacent port, and steam "follows full-stroke." When they are widely separated, but a small proportion of the stroke is made before the port is covered and the steam cut off. The Meyer valve, the Rider, and others, some of which are to be presently studied, are illustrations of this class.

In another form, illustrated by the Gonzenbach gear, Fig. 104, the valve-chest is double, the chamber nearest the cylinder enclosing the main valve, the second, usually superposed, containing the cut-off valve, which is generally of the gridiron form, covering a seat in which are cut the several ports corresponding to the bars of the valve, and alternately covered and uncovered by the latter. This valve is used for fairly high ratios of expansion, and is most prompt and efficient in its operation when cutting off at about one-quarter stroke. It is obvious that the valve must open its port at the same value of crank-angle near the end of stroke as that at which the closure occurs in the earlier half-stroke. The main valve must therefore be given sufficient lap to cover the steam-port before this reopening by the cut-off valve can take place; thus, if the cut-off valve be set to close its ports at one-fourth stroke, the main valve should close the port as early as at three-quarters stroke. The cut-off valve is usually so arranged that its ports shall be wide open at its middle-throw, coinciding with those in the seat. When it has moved, either way, the breadth of the port, cut-off occurs. The point of cut-off is then determined by the throw of the valve; the longer its stroke, the higher the ratio of expansion. It is therefore often made adjustable as to stroke in order that a variable expansion may be secured. The eccentric driving it is set without lead, so as to move it past mid-stroke at the beginning of the piston-stroke. A section of such a valve is shown in Fig. 104. *A*, *A*

FIG. 104.—PLATE CUT-OFF.

are ports in the valve-seat. B, B are ports of the same size in the valve. When the valve is in its middle position, and the piston at one end of its stroke, the ports B are exactly opposite to the ports A, which are then wide open.

The point of cut-off being given, the following are methods of finding the requisite proportion of breadth of openings to half-throw:

About a centre, O, describe a quadrant, DE, and draw a radius, OD, to represent the half-stroke of the piston, and take the point R to represent the point at which the steam is to be cut off. From R draw RG perpendicular to DO, cutting the circle in G. Then, since ordinates parallel to GR represent the proportion of opening as the crank swings through the quadrant, as already seen in the preceding article,

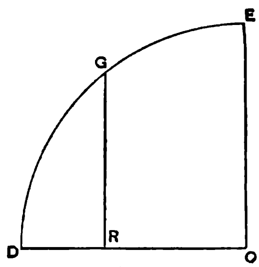

FIG. 105.—VALVE-OPENING.

$$\frac{\text{Breadth of openings}}{\text{Half-throw of valve}} = \frac{\overline{RG}}{\overline{OE}}.$$

The travel can also be readily calculated. Let the ratio of expansion

$$\frac{2\,OD}{DR} = r;$$

then, from the equation of the circle,

$$y^2 = 2Rx - x^2; \qquad y = \sqrt{(2Rx - x^2)};$$

$$y = RG; \qquad x = \overline{DR} = \frac{2}{r}.$$

Take $R = DO = 1$, and

$$\frac{\overline{RG}}{\overline{OD}} = y = \sqrt{\left(\frac{4}{r} - \frac{4}{r^2}\right)} = \sqrt{\frac{4(r-1)}{r^2}};$$

the angularity of the rods being neglected.

This case is very beautifully treated by Zeuner's graphical method, which has one of its most useful applications in the treatment of the system of valve-motions of which Meyer's gear is the type.

The motion of the Thompson cut-off valve, relatively to the main, is similar to that of the Gonzenbach cut-off block on its fixed seat.

In the operation of the Meyer system, the cut-off valve worked by an independent eccentric, the latter is commonly set opposite to or with the crank, and at right angles to the main eccentric; the point of cut-off being varied by altering the lap, spreading more or less the two blocks of which the valve is composed, by means of a right-and-left screw. Other methods are, however, practicable, and such effects are secured either by altering lap, as just described; by altering the lead of the eccentric, the lap being constant; by the use of a link to vary the stroke of the valve; or by varying its travel by other and special expedients. In some cases, as in some marine engines, the expansion-valve is driven by a system of levers actuated by the piston-rod or cross-head.

Where a cut-off valve rides on the back of the main, their relative motion is obtained thus:

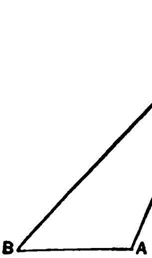

FIG. 106.—THE CUT-OFF VALVE.

In the figure, let AB and AC be the eccentric radii of the main and the cut-off valve, respectively; then their *relative* motion is that which would be obtained were the first at rest, and the second driven by an eccentric, of which the action-radius is BC.

With the Meyer system, the opening of the steam-ports and the opening and closing of the exhaust-ports are secured by the action of the main or distribution valve, and are not at all affected by that of the expansion-valve, and these movements are to be traced by means of the main-valve diagram. It must be made certain, however, that the closing of the port by the main valve takes place before the reopening of the port by the expansion-

valve can take place, this latter event occurring at a point just as far from the end of the stroke as is the point of cut-off from the beginning. Thus, if the expansion-valve cuts off steam at one-third stroke, the main valve must have lap and lead enough to close the port at or before two-thirds stroke. The expansion-valve cut off the steam entirely independently, and its action is to be studied by means of its own circle and diagram. It must have sufficient lead, however, to permit the opening of the steam-port by the main valve to occur at the right time. The two plates of the expansion-valve should never be so far separated as to cover the ports at the beginning of the stroke. The extent of port-opening given by the expansion-valve is measured on the diagram obtained by the production of the resultant circle.

When using the Meyer gear, the custom has been, with marine engines especially, to place the cut-off eccentric either with the crank or opposite it; but it is perfectly practicable to shift it somewhat, and thus secure reduced travel of the cut-off blocks, when shifted to alter the expansion. The range adopted is usually from zero to the point of cut-off of the main valve. The system is not a quick-acting or a satisfactory one, except at about a ratio of expansion of four.

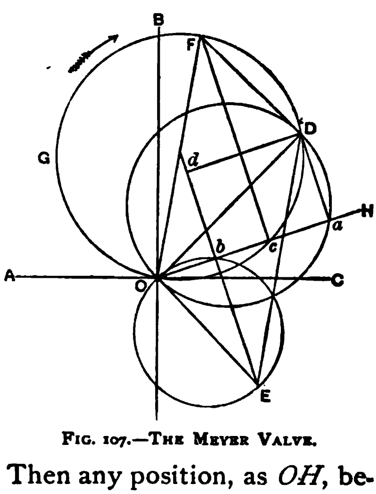

FIG. 107.—THE MEYER VALVE.

The action of the Meyer expansion-valve and its motion relatively to the main valve are very easily and simply exhibited by means of the Zeuner diagram. Thus, in the figure, let *AOC* represent the path of the piston; let *OB* be the normal position of the crank when the main valve is in the middle of its stroke if without lap or lead; and let *OD* and *OE* be the positions of the eccentrics and half-travel of the two valves at the beginning of the engine-stroke. Then any position, as *OH*, be-

ing assumed for the crank, the chords, Oa, Ob, will measure the distances moved over from their middle position by the main and the cut-off valves, respectively. The motion of the one relatively to the other is the difference of these quantities, or $Oa - Ob$.

This quantity and the relative motion of the two valves may be exhibited by the use of a single valve-circle, $DFGO$, constructed thus:

Draw DE and make OF and FD parallel, respectively, to DE and OE. Then the circle, $DFGO$, constructed on OF as a diameter, will be a valve-circle exhibiting the movement of the cut-off valve on the back of the main valve, and their action relatively to each other. For, drawing Ed, Dd, making the angle at d a right angle, since DE is equal and parallel to OF, the right-angled triangles OFc and EDd are equal, $Oc = ab = Dd$, $aO - ac = aO - Ob = Oc = ab$, and Oc measures the motion of the one valve relatively to the other; the middle position of the cut-off valve on the main valve being taken as the base position gives the measure, OI or OJ, of the steam-lap on the main valve when it is to effect a cut-off at OB.

To measure the travel and angular advance of the cut-off eccentric, draw Ea, EB; then EOA is the angle by which the cut-off eccentric leads the main eccentric; the former being set with the crank. On EAO draw a circle, $ANRO$, the cut-off valve circle. Then OR is the distance travelled by the cut-off valve when the crank is at OE; EK thus measures the travel of the cut-off valve to cut off steam when the crank is at OE; ON is the travel for a cut-off at OM.

Many modifications of the Meyer type of expansion-gear are known, all of which may be similarly treated. One of the best known is the Rider system, in which the expansion-valve is a cylindrical piece fitted to a cylindrical seat on the back of the main valve and arranged to revolve about its axis. In plan the valve is trapezoidal, and the ports in the main valve are set at angle and parallel with the external inclined edges of the expansion-valve, the motion of the valve being parallel with the two parallel sides of the valve. When the valve is

turned by the governor or by hand, the lap is increased or diminished, and the effect is precisely the same as when the two parts of the Meyer valve are varied in their distance.

To determine the motion of an auxiliary cut-off valve on the back of the main slide, in Fig. 108, we may also proceed thus: Draw *AH* proportional to the travel of the main valve; *AO*, *OH* being the positions of the crank when on the centre;

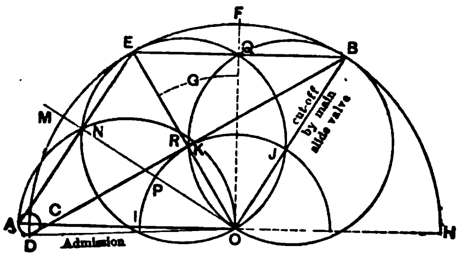

Fig. 108.—Motion of Expansion-valve.

and make *OB* its position when cut-off would take place by the action of the main valve, and *OD* that at which steam enters; *AOD* being thus the angular measure of the lead, and *ADC* the "lead-circle." Bisecting *BOD* and drawing *OE*, this gives the position of the crank at the extreme throw of the eccentric; and *EOH* will be the angular advance of the crank. Taking *OK*, the half-travel of the valve, as a radius, and drawing the "lap-circle," *IKJ*, we find all critical points.

In designing the Meyer or similar forms of expansion-gear, the same principles must be observed in the proportioning of the main valve as when planning the single-slide valve, as already outlined. Further, as it is necessary to see that the main valve has sufficient lap to suppress admission before the cut-off valve reopens its port toward the end of the stroke, and common arrangement fixes the cut-off by the expansion-valve at one-fourth to three-eighths stroke, its best work being done usually at about the earlier of these points, every expansion-valve should be given such port-area as will prevent retardation of the current passing through the main valve, such relative proportions as will insure a sharp cut-off, and as much

range of expansion as is practicable. A good system of variation of the position of the blocks is essential, whether by hand or by the governor.

The design of an expansion-gear on the Meyer type has been seen to be reducible to the single-valve system, so far as proportioning goes, by means of the Zeuner diagram or an equivalent method. The following design, in greater detail, further illustrates this problem as applied to the form in which the expansion-valve is superposed upon, and in the same valve-chest, with the main valve. In the figure let *AA* and *BB* represent the main and the cut-off valves, respectively. *CD* is the

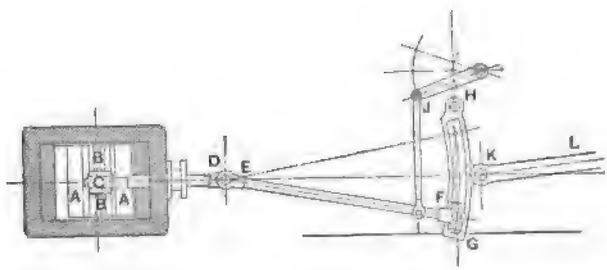

FIG. 109. INDEPENDENT EXPANSION-VALVE AND LINK.

valve-stem of the cut-off valve; *EF* is the connecting-rod, by means of which it is united to the link-block, *F*, sliding in a half-link, *GH*, swinging about a centre, *H*, in such manner that the block may give maximum movement to the valve or may hold it at rest, as it is lowered by means of the suspension-rod, *IJ*, to the bottom, or raised to the top. *KL* is the eccentric rod giving motion to the link; to which it is attached at *K*. The dotted lines indicate the varying positions of the several parts and the centre-line of the gear.

Here the main-valve operates precisely as if no cut-off valve were attached, and its lap and lead determine the minimum ratio of expansion. The device added is evidently a supplementary valve, of which the travel may be altered as required to produce a variable expansion. Its earliest cut-off should be at zero if possible, in order that the steam may be suppressed

entirely should all work be accidentally or purposely thrown off. To proportion the valve, proceed thus:

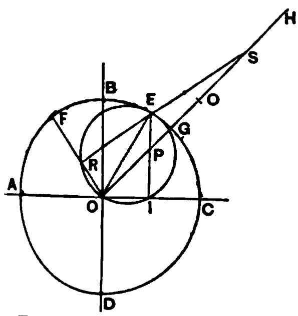

FIG. 110. THE EXPANSION-VALVE.

In the figure, let AC and BD be two centre lines passing through the shaft line O. Let E be the centre of the main eccentric when the crank is at A; and make OB and OF the positions of the latter at earliest and latest points of cut-off. Describe, with the radius OE, the valve-circle $AFECD$, the radius OE representing the half-travel of the main valve. On the arc BEC set off from E a distance, EG, equal to the minimum movement of the expansion-valve on the back of the main valve, thus locating the position of the expansion-eccentric, whose centre must be in a radius, OH, or "line of centre," passing through G; OI is the lap of the main valve, and EI is normal to OC. The point, P, at which this line intersects OH, locates the position and determines the throw of the expansion-eccentric, which would give the valve a throw, or travel, such as would cut off steam at the beginning of the stroke. Similarly a line RES, drawn from R through E and cutting OH at S, gives the position of the centre of an eccentric which would allow a travel such as would give maximum expansion desired, the cut-off occurring when the crank is at OF. Now assume a link to be introduced between the eccentric and the valve. The proportions of the link and method of suspension should be such that, a single cut-off eccentric operating it, the travel of the expansion-valve should be variable between the maximum and minimum just ascertained. An eccentric of any convenient throw may be employed to move the link, if attached properly; but the lead must be that shown by the line of centres, OH. The valve itself must have a width such that, with minimum throw, the steam ports should be covered at the position OF; and then at maximum throw the ports will be covered at all times during the stroke.

To lay out the link, let AB be the centre line of the expansion-valve and stem, and passing through the shaft-centre at O, the head of the valve-stem being at C. Draw DE perpendicular to AB, and at the proposed position of the link, and so locate the point of suspension E, and the ends F and G, of the link that

$$EG : EF : FG \text{ (Fig. 111) } :: HS : HP : PS \text{ (Fig. 110)},$$

and the action of the link will be what is intended, the points F and G giving maximum and minimum throw respectively, and intermediate points in the link all intermediate throws.

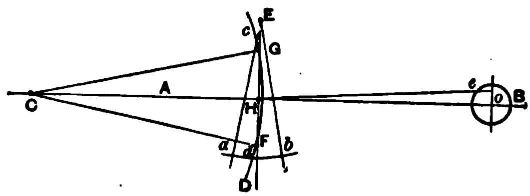

FIG. 111. THE EXPANSION-LINK.

The eccentric at O, if coupled to the link at H, should have a throw such as to give the proper range of motion of that point in the link. CD, CG are the extreme positions of the radius rod, and the angle aEb is the swing of the link. The line of its slot is cHd; eH is the eccentric rod. The block may be moved in the link either by hand or by the governor. In the latter case especial care should be taken to secure ample area and good lubrication of the block and the link.

The independent expansion-valve in a separate steam-chest, as that of Meyer's, is still more easily designed. It is necessary to see that the blocks are so proportioned that when closely brought together they will not close either steam-port at either end of their range, and that when farthest separated they will not allow steam to enter within the intended range of suppression; and their adjustment will, if the throw be right, effect all that is desired in varying the ratio of expansion.

The motion of the expansion-valves of the Meyer valve-gear

is well shown by the diagram below, as drawn by Uhland's method. The movement of the main valve follows the dotted curve, of which the ordinates measure the movement of the piston, and the abscissas the valve-travel.

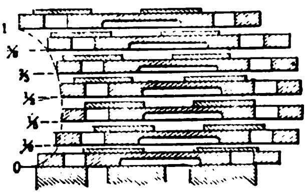

FIG. 112.—MEYER VALVE-MOTION.

The cut-off plates have a relative motion on the back of the main valve which is given in the figure for each step in the movement of the piston and of the valve. The peculiarly excellent action of this valve for ratios of expansion between 3 and 5 or 6 is best exhibited by making a diagram of speeds.

Proportions which are generally considered satisfactory for the Meyer valve and its movement are as follows, the diameter of cylinder being taken as the unit of measurement, and as 1000:

Diameter of cylinder	1000
External lap of main valve	18
Internal " " " "	8
Angular advance of expansion-eccentric	15°
" " " main eccentric	68°
Eccentric-radius (expansion)	89
" (main)	100
Steam-port, width	70
Bridge, width	85
Exhaust-port, width	100
Valve-port (cut-off)	75
Length plates	230

A very simple form of expansion-valve, closely related to the Meyer, is constructed with a main valve similar to that of the latter and an expansion-valve in a single plate, or in two parts, according to its position relatively to its ports, lying loosely on the main valve. Adjustable stops are fitted inside the steam-chest, against which the expansion-valve strikes as it approaches either end of its travel and, by the relative motion

of the two valves thus produced, the steam-ports are closed. The point of cut-off is determined by the extent to which these stops are set out to meet the approaching valve. They may be adjusted either by hand or by the governor.

The lateral pressures on valve-stem guides, as on other crossheads, depends on the ratio of lengths of link or of rod and swing of its head. Calling it Q, and measuring r and L in similar units,

$$Q = P\frac{r}{L}.$$

The mean value of the tan $\theta =$ tan DAB, is $BD \div AB$. But, $BD =$ area $EFG \div CG$. $EFG = \frac{1}{2}EFGH \div EG = \frac{\pi}{4} \times \frac{S^2}{2} \div S$; if $S =$ stroke of engine or diameter EG. Then if L is the length, AD, of the rod, and r is the radius, CG, the mean pressure is

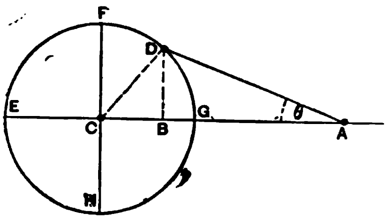

FIG. 113.—CRANK-MOTIONS.

$$T \tan \theta = T\frac{BD}{AB} = T\frac{\frac{1}{4}\pi r}{\sqrt{L^2 - \frac{\pi^2 r^2}{16}}} = T\frac{0.7854r}{\sqrt{L^2 - 0.617r^2}}. \quad (1)$$

The ratio of the lateral and direct forces, θ, along BD and P on AB, is

$$\frac{Q}{P} = \frac{BD}{AB} = \tan \theta.$$

The ratio of the stress, T_1, along the rod AD, to the component P is

$$\frac{T}{P} = \frac{AD}{AB} = \frac{L}{\sqrt{L^2 - 0.617r^2}}. \quad . \quad . \quad . \quad . \quad (2)$$

If f be the coefficient of friction, the efficiency of this sliding piece will be e, and the counter-efficiency c,

$$e = \frac{P}{P + fQ}; \quad c = 1 + \frac{fQ}{P}. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

56. **The Cornish or Plugtree Valve-gear** is that used in the later pumping-engines of Newcomen, and in those of Watt and later builders of engines without crank and fly-wheel. In this system the valve-motion is actuated by the beam, the air-pump connections, or other part of the mechanism, moving synchronously with the piston. The distinguishing characteristic of this engine, there being no fly-wheel, is that all movements, of whatever part of the system, must commence with the beginning of the stroke of the engine and must, at all times, be proportional in speed and in space traversed to the velocities and distances described by the piston, and must cease with termination of the piston-stroke. No lead can be secured by this movement; but the point of cut-off can be readily varied. The use of the "cataract," however, in the Cornish engine, enables the time and number of strokes to be fixed at will and varied at pleasure.

The design of this form of gearing can be studied in the older works on the steam-engine; it is now seldom employed, and has comparatively little interest, except historically. A modification of this system, however, in which, while the movement of the gear is nearly or exactly coincident with that of the piston, the actuating element is an eccentric, instead of a beam-motion, has been introduced more recently, and is of real importance.

57. **The Stevens Valve-gear,** the joint invention of Messrs. Robert L. and Francis B. Stevens, in 1848 (see Fig. 110, Part I), is in very common use on the engines of river steamers in the United States, where the beam-engine is employed. Here one eccentric is used to operate the exhaust-valve, and is set precisely as if to work the plain slide-valve; while an independent eccentric is employed to actuate the steam-valves, and is given increased throw and restricted range of action in contact with the valve-connections, thus securing the effect of lap on the slide-valve and a moderate amount of expansion.

This action is graphically illustrated thus:

Let the eccentric-radius be represented by AO, the full throw

of valve-motion by AB, and let OC be the extent of movement required to open the valve fully to steam and, returning, CO the travel while seating the valve. Let it be required that the valve shall be held open while the crank is traversing an arc equal to Oab; the steam-port must then begin to open when the eccentric is at Oa, and must close at Ob. The lift of valve being proportional to Cc, the eccentric must commence lifting the valve after a rise Oc, and continue the operation through cC; it must therefore be so set that it shall traverse the arc aCb while opening and closing the valve, must be set ahead so as to begin raising it at a, the crank just passing the centre, and AOa, or BOb, measures the angle by which it is to be set ahead on the shaft.

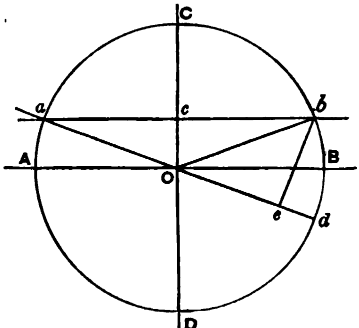

FIG. 114.—STEVENS VALVE-GEAR.

Let fall the perpendicular, be, on aO produced to d, and the ratio $ae \div ad$ measured the proportion of its stroke traversed by the piston up to the point of cut-off. Practical difficulties of application obviously preclude the use of this gear for high ratios of expansion, and it is rarely set to cut off at less than six-tenths stroke.

In laying down the Stevens or any other valve-gear operating the valves by a system of wipers, toes, and lifters, the designer should endeavor to so proportion it and to so arrange the sizes and the curves of the wipers that the valve should be started off its seat by the action of the heel of the wiper, then, contact being suddenly transferred toward the point of the toe, it should be quickly lifted to the height of its rise. By this arrangement the very considerable resistance usually met with at the start is overcome with best advantage and yet a prompt and full opening is insured. It will be found necessary to adopt a longer and straighter wiper for the steam- than for the exhaust-valve, and a longer throw for the exhaust- than for the steam-eccentric. The working arc of contact traversed by the

exhaust is much greater than that of the steam-wiper. Smooth and easy curves are desirable on their working edges.

58. **The Sickels Valve-gear,** invented by Frederick E. Sickels in 1841, was the first practically successful "drop cut-off." It has been extensively employed on American beam-engines as used on board river and inland steamers. It consists of a set of steam-valves, usually of the "double poppet" variety, operated by the eccentric, which also actuates the exhaust-valves, but connected in such manner that the steam-valve, rising from its seat and moving synchronously with the exhaust-valve, may be detached at any desired point and allowed to drop upon its seat, suppressing the entering current of steam and thus effecting the "cut-off." This disengagement is produced automatically, by a wedge or stop against which the catch holding the valve to the "lifter" strikes as it rises and thus liberates the valves. The position of this stop determines the point of detachment and cut-off; and this is either fixed or is adjustable automatically or by hand, the former system being adopted on stationary engines, where the governor may be arranged to do this duty, the latter method being the usual one in other cases.

The *dash-pot* is a distinguishing and essential feature of this invention. When the fall of the valve occurs, under the impulsion of its own weight and the partly unbalanced steam-pressure, it is certain, if unchecked, to strike a heavy blow on its seat, injurying both surfaces and causing disagreeable jar and noise. To prevent this, there is attached to some convenient part of the falling valve attachments a piston or plunger which works into a cylinder partly filled with water or oil, or sometimes merely enclosing air. As the valve approaches its seat, the piston or plunger strikes the surface of the liquid or compresses the air, thus preventing or reducing the shock without seriously interfering with the efficiency of the cut-off mechanism. Provision being made for readily adjusting the depth of the oil or the extent of compression of the air, the effectiveness of the dash-pot can always be insured, and the device gives a noiseless, prompt, and complete suppression.

As above described, however, it is evident that cut-off cannot take place beyond half-stroke. The eccentric, controlled by the requirements of the exhaust side, begins raising both steam- and exhaust-valves at half-throw, reaches the end of its travel at half-stroke, nearly, a little variation being produced by the lead, and then commences lowering the valve. If the stop has not been reached at half-stroke, cut-off cannot take place and the engine "follows full-stroke." But in some cases this disadvantage is overcome by the inventor in a very ingenious manner, by the use of the "beam-motion," as in the accompanying skeleton-drawing.

Fig. 115 is a diagram of Sickel's cut-off, in which *A*, *A*, is the steam-valve; *B*, *B*, valve-stem; *C*, dash-pot, filled up to the line 1, 2; *D*, plunger in the dash-pot; *E*, stuffing-box; *x*, hole in the plunger, *D*, to allow water to enter when the plunger strikes; *b*, a rod communicating motion to the *wiper*, *F*, which, in this arrangement, trips the valve; *h*, a rod or any actuated part having motion coincident with the piston. The motion of *h* occurs through the vertical rod, having *c* as a centre to *b*, and thence to *F*. The valve is lifted by the eccentric, but before it reaches its seat, the wiper *F*, which vibrates back and forth, strikes the clutch, and detaches the valve which is attached to the plunger *D*, working in the dash-pot *C*. Before the valve reaches its seat, the plunger *D* strikes the water in the dash-pot *C*, called the secondary reservoir, the water escaping into the cavity *x*, and also into the upper or primary reservoir.

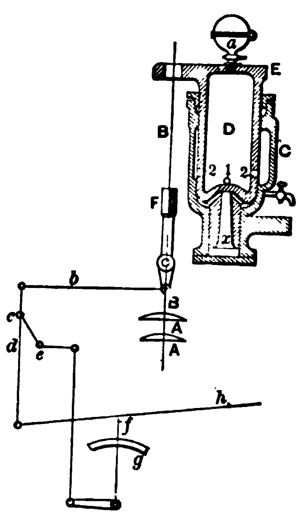

FIG. 115.—DASH-POT.

The adjustment of the cut-off is effected by moving the handle *f* on the arc *g*, moving the centre *c* to one side or the other, *e* remaining fixed, and therefore giving the wiper a greater or less travel before striking. The handle *f* can be placed so that

the valve will not be tripped at all, or so that the valve will not lift at all, being exactly in the vertical position when the lifter commences rising. The engine can be thus stopped.

The design and construction of the lifting system for the Sickels gear is substantially identical with the exhaust side of the Stevens, and the principles which should control the form and proportions of its details are the same. The whole apparatus should be designed as if it were intended to work independently of the cut-off gear, but always at full-stroke. The valves should be started from their seats easily, then thrown up to the full height of their lift by the action of wipers so formed and of such size that the ascent shall be amply sufficient to give promptly a full opening, and then to let the valves down upon their seats quickly, but seating them without shock. The addition of the cut-off gear to such a system is then a very simple matter.

59. **The Corliss Valve-gear** is a type by itself, as is the engine of which it forms a part. Including four valves, a steam- and a separate exhaust-valve at each of the steam-cylinders, it provides independent steam- and exhaust-ports at each end, and this permits them to be made very short and of very small volume, thus evading loss by excessive clearance. It also reduces the loss by condensation in the passages to a small amount, as the entering steam is not compelled to traverse passages previously cooled by the exhaust. Being a detachable, or drop, cut-off, the steam-pressure and temperature is kept very nearly at the maximum throughout the period of admission, and, the exhaust-valves being capable of adjustment for best effect, the back-pressure line exhibits minimum and nearly uniform pressure and the cycle is made substantially that of Carnot, that of maximum efficiency.

An important characteristic of the Corliss gear is the interesting and ingenious kinematic system which permits the valve-motion to act rapidly while opening or closing a port and yet to move very slowly when the port is well open, thus securing prompt and full admission, ample port-opening, and very slight frictional resistance. The peculiar form of valve,

also, permits easy operation, and it is thus made a fair equivalent for the balanced valve. The turned valve, the bored seat, and the simple method of connection give an easy and cheap construction. The system, as a whole, is found to be extraordinarily satisfactory and economical.

The general arrangement of the Corliss gear is seen in the accompanying figure, and a study of it and its details will show plainly its excellence in these characteristics, giving quick closing of the steam-valve, very small clearance and "dead-space," a cut-off regulated by the governor, independence of the several valves, and complete adjustability.

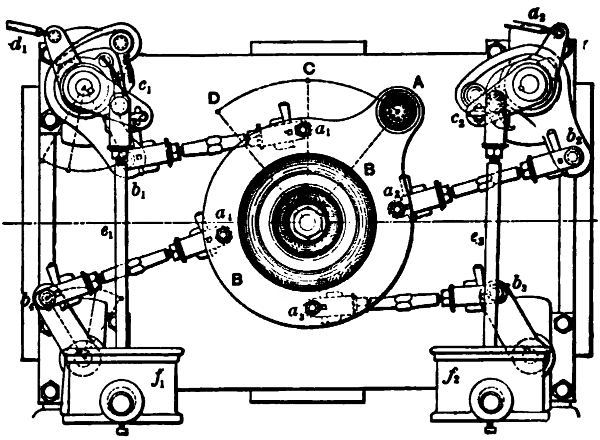

FIG. 116.—THE CORLISS VALVE-GEAR.

There are many modifications in detail of this system, but that here illustrated may be taken as a good example. The eccentric-rod is attached to a pin, A, on the "wrist-plate," BB, and gives it a motion similar, as to time, to that of the ordinary simple three-ported slide-valve, vibrating through the arc, ACD. Points, a_1, a_2, a_3, a_4, are selected on the face of this wrist-plate such that valve-rods, a_1b_1, a_2b_2, a_3b_3, a_4b_4, being attached to the rock-arms, b_1, b_2, b_3, b_4, the extreme of the vibration of the wrist-plate, occurring at half-stroke, throws the rock-arm into such a relation with the connecting valve-rod as will secure a minimum motion in that stage of operation for all

points. At the same time, the connections are so related that, at the commencement of the stroke, the eccentric then in mid-throw, the several rods and radii are disposed in such manner as to cause motion of maximum rapidity, opening a pair of steam- and exhaust-valves and closing those at opposite ends of the cylinder early and quickly. Thus the port-opening takes place rapidly, the valves move rapidly while opening and very slowly afterward, and the system constitutes an example of a remarkably efficient valve-gear.

The detaching apparatus, c_1c_2, is variously designed; but it commonly consists of a cam, turning on the valve-spindle, and throwing out the latch by means of which the connection is established between the valve-arm and valve-link. By turning this cam an earlier or a later release is effected as the latch, swinging with the rock-arm, comes into earlier or later contact with it. The regulating mechanism turns the cam. This may be done by hand; but it is almost invariably, on stationary engines, actuated by rods, d_1, d_2, connecting with the governor in such manner that the variation of speed determines the point of cut-off. Another pair of rods, e_1, e_2, connect the valve-arm with the dash-pots, f_1, f_2.

The proportions of this system of valve-gear are best determined by laying it down on the drawing-board and adjusting its parts until the desired steam-distribution is secured. The fact that the release of the steam-valve cannot take place beyond about half-stroke makes it necessary to restrict the lead to the least practicable value, as every increase of lead reduces the range of cut-off. It is partly for this reason, also, that compression is rarely adopted with this engine. It is perfectly practicable, however, to employ separate eccentrics for the steam- and exhaust-valves, setting the former next the crank and thus to secure a range of cut-off extending from the beginning nearly to the end of the stroke.*

The Corliss valve-gear, as ordinarily constructed, is, as al-

* For full descriptions of a remarkable variety of Corliss valve-gears, see Uhland's "Corliss Engines;" London and New York, 1879.

ready seen, not adapted to cases of variable load which demand a range of cut-off beyond about forty or forty-five per cent of the stroke. A separate eccentric and resort to the plug-tree type, as illustrated by Greene's engine, is necessary to secure a wider range of expansion. It is because of this difficulty in the common Corliss engine that it is so generally given no steam-lead.

Another notable point in the design of this gear is that which compels the use of a different form of valve for steam and exhaust. The exhaust-valve is necessarily placed in a box so shaped that the pressure of the steam within the cylinder shall hold it on its seat; its seat is therefore on the inner surface of this valve-box, and over a secondary exhaust-port, instead of on the cylinder itself; and the valve, Figs. 34, 35, Part I, meets this need. The steam-valve is also so made and so set that, when closed, it may have ample lap to insure against leakage; but this lap must be taken up before the port begins to open, and the available range of cut-off is thus sensibly restricted. The adoption of independent movements for steam- and exhaust-valves, to avoid this difficulty, has been more usual in Europe than in the United States. It is also thus made practicable to secure some compression with its attendant mechanical and economical advantages.

The Corliss dash-pot, as designed for the Watts-Campbell engine, is seen in the figure. The engine-valve being open and the plunger raised, a vacuum in the chamber at the upper part of the central cylinder causes a quick fall, and cushioning takes place in the annular space at the bottom. The two adjustable plugs serve to make the motion as rapid or as slow as may be desired, and the check-valve at the top of the central cylinder discharges any air entering by leakage or through the screw-plug. High efficiency with

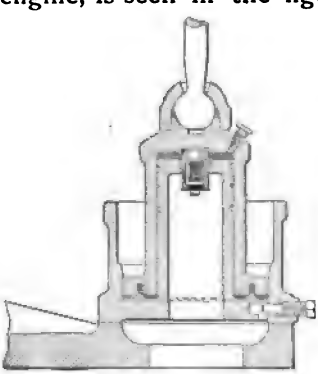

Fig. 117.—Dash-pot.

this apparatus is indicated by a prompt and rapid drop, closing

the valve, and freedom from shock when stopping. Leather cushions under the plunger are additional safeguards.

In laying out a design for a Corliss valve-motion, the main points to be secured are: such an adjustment of the relative positions and motions of wrist-plate pins and valve-arms as will give the desired rapid opening of port and later slow movement which is a characteristic of this gear; maximum range of expansion consistent with a good steam-distribution in other respects; good action of the governor and a trip-motion that shall be safe and exact without affecting the sensitiveness of the governor; and such relative motion of the valve over its ports as will insure permanent tightness of valve and accuracy of movement. The setting of the steam-valve should be made as for full-stroke, and of the exhaust as for the exhaust side of the plain slide-valve, with the qualification, however, that the lead should not be so great as to restrict sensibly the range of expansion.

The Corliss movement, through the action of its peculiar kinematic train and the valve-levers, as shown, gives, as already stated, a modification of the motion of the valves producing a more prompt opening and closing, and slower movement while open or closed; while less work is thrown upon the engine and the governor than with the older systems of operation. The very long and narrow ports aid in this work by permitting full opening at a very early stage in the travel of the valve. This modification of the motion of the steam-valve is illustrated in the accompanying curve; in which the ordinates exhibit the travel of the valve, and the abscissas the corresponding motion of the piston; in the order of letters and figures, respectively. The lower scale marks the piston-movement in the forward stroke, and the upper line of figures exhibits its progress on the return; while the relative motion of the valve is shown by the curve *ABCDEG* and by *GHIJKA*.

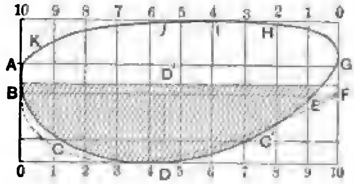

FIG. 118.—CORLISS MOTION.

When the piston is on the centre, at *B*, the valve has already moved sufficiently to open the port and give the desired lead, *AB*. When the piston nearly reaches its half-travel mark, at *D*, the valve is wide open and then it gradually closes. But the port is of the width *AC*, and is wide open when the piston reaches the point *C*, at one tenth its stroke in this case. If the valve is tripped after reaching this point and before reaching *D*, it will always have given maximum steam-opening. At *C* the closing of the valve begins to throttle the steam, again. As the valve begins to retreat at the line *DD'*, disengagement cannot take place beyond that point, and the steam will follow nearly full-stroke, if the cut-off is not effected earlier. The introduction of angular advance of eccentric and lead of valve is seen to correspondingly shorten the period available for variation of expansion, in the case here shown, restricting this range to *OD* and the final closing of the undetached valve to the point *E*. Assuming the latter case, the motion of the valve during the return-stroke is seen on *GHJAB*, the greater part of the movement taking place at either end of the stroke, and the valve moving but slightly in the intermediate period.

The proportions of the parts and the general design, itself, of this valve-gear are subject to considerable variation in the various engines of the inventor and of other makers; so that the limit of expansion and the relative velocities of the valves in the different parts of their paths are somewhat variable; but in all cases the advantages here illustrated appear in greater or less degree. The dotted line in the last diagram illustrates an extreme case.

The next illustration exhibits the working of a Corliss compound engine described by Uhland;* *A* and *B* showing the movement of the steam- and exhaust-valves of the high-pressure, and *C* and *D* those of the low-pressure, engine; the working edges of the valve being also shown at the beginning of the stroke. Separate eccentrics are here used for the steam-

* Corliss Engines; p. 55.

and exhaust-valves; and thus the range of cut-off increased, as shown by the extent of the hatched section on the steam-valve diagram. Since the valve must, in this case, be tripped, the return-motion line represents only what would occur were non-

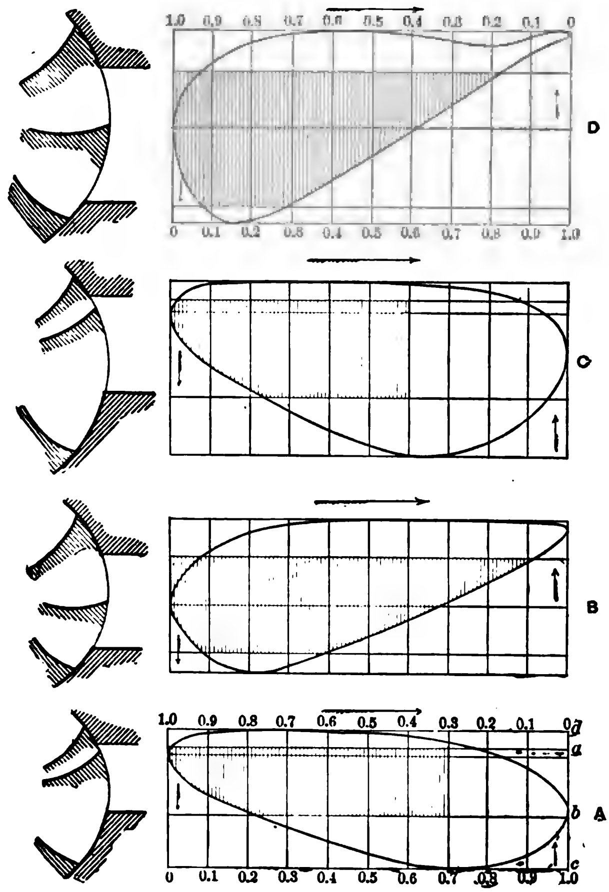

FIG. 119.—COMPOUND CORLISS ENGINE.

expansive working practicable. The breadth of port is represented by the distance *ab*; while the limits of valve-travel are *c* and *d*.

The peculiar movement due to the Corliss system of ad-

justment of the valve-connections is here clearly seen in the deviation of these curves from the ellipse.

The Corliss movement is well exhibited by the polar con-

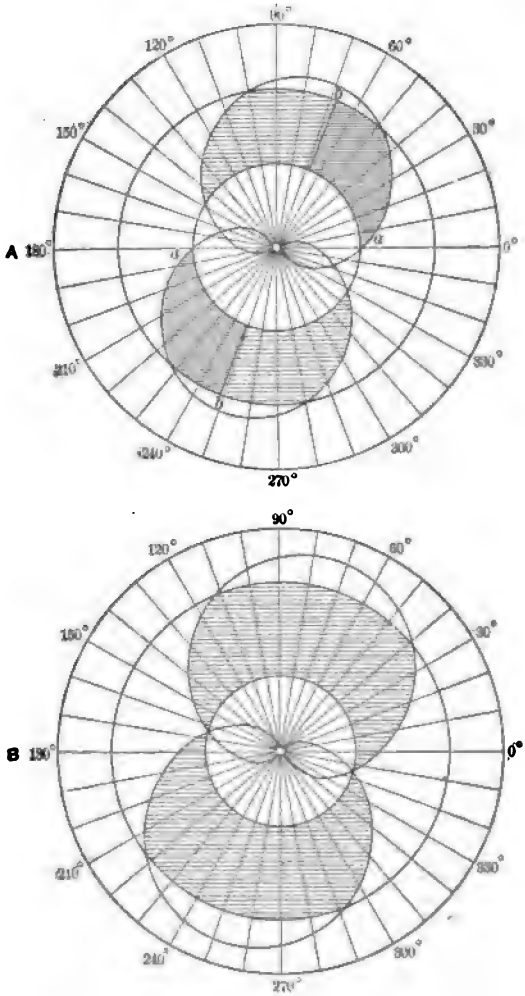

FIG. 120.—POLAR CONSTRUCTION, CORLISS MOTION.

struction seen in Fig. 120, as illustrating the gear brought out by Mr. Corliss in 1851, with a large wrist-plate driving the several rock-shaft or valve arms. Diagram *A* shows the motion of

the steam- and *B* that of the exhaust-valves, and the outer circle marking the angular movement of the crank, the inner concentric circles the limits of the port; while the smaller, limaçonoid, curves exhibit the valve-travel and correspond with the inner twin-circles in the Zeuner diagram for the common simple eccentric motion. On these small twin-curves, the portion of each external radius intercepted by its circumference measures, as in the Zeuner diagram, the corresponding departure of the valve from its initial position. The radius of the inner circle measures lap.

In *A* are shown the points of maximum range and cut-off, *ab*, for the steam-valves, those points at which the valve must be tripped, if at all. The darkly hatched portions enclosed between the valve-circles and the port-circles exhibit the progressive opening and travel to that limit. The variations of the curves from the circular form show the kinematic difference between the Corliss valve-movement and the simple eccentric and slide-valve motion, and exhibit the comparatively rapid opening of the steam-valve, and the quick closing, as well as opening, of the exhaust. The outward movement along the line of the valve-circle, and its return, are seen to take place much more sharply across the band representing the port-space than in the ordinary case; this fact gauging the improvement in this point effected by the introduction of this type of valve-gear.

It is also seen that the port, even here, is not fully opened at any short cut-off. The narrower the port in this line, the area being fixed, the less does this defect appear. The lead, as measured at the diameter 0°–180°, is seen to be inconsiderable on the steam side but much greater on the exhaust, a method of setting, in a certain sense, necessary to secure a good range of adjustability of the cut-off. On the exhaust side, *B*, the lead is nearly one third the breadth of port and the port is wide open at a crank-angle of 35°, and at about one tenth the stroke. The port is again closed at 165°, just inside full return-stroke, and about 15° before the opening of the steam-valve, giving slight compression.

Mr. Henthorn advises the following adjustments of valves:*

Diam. Cyl. Inches.	Steam-lap.	Exhaust-opening.
	Wrist-plate at centre of motion.	
16″	$\frac{5}{16}$″	$\frac{1}{32}$
20	$\frac{3}{8}$	$\frac{1}{16}$
24	$\frac{7}{16}$	$\frac{3}{32}$
28	"	"
32	$\frac{1}{2}$	$\frac{1}{8}$
36	"	"
42	$\frac{9}{16}$	$\frac{3}{16}$

60. The Greene Valve-gear is a good illustration of a modification of a plugtree valve-gear, with a system including an eccentric. As has been seen, the operation of both steam-

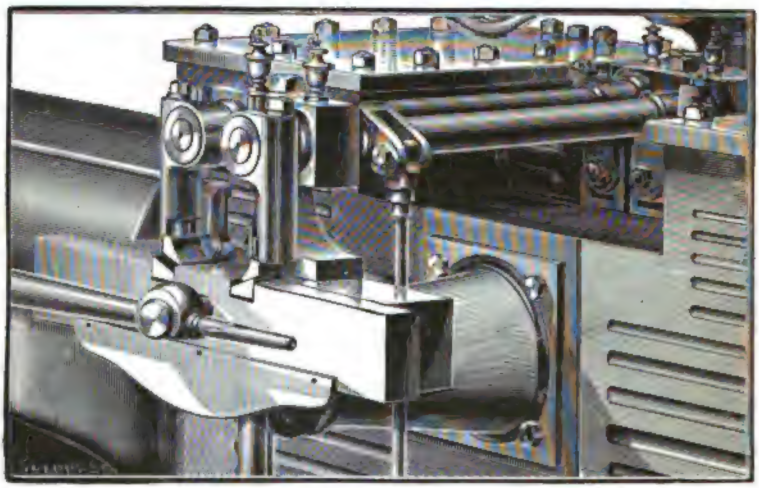

FIG. 121.—THE GREENE VALVE-GEAR.

and exhaust-valves by the same eccentric-motion, in consequence of the fact that the exhaust valve-motion must control the position of the eccentric, does not permit a drop cut-off mechanism to be so arranged as to close the port after passing

* The Corliss Engine; p. 25.

beyond half-stroke; and this restricted range of expansion is, in some cases, a defect. To avoid this objection, Mr. Greene has used either a separate eccentric, or, in later engines, two sets of connections actuated by a single eccentric, the one receiving motion synchronously with that of the piston, the other, connected to the eccentric at right-angles to the former, giving the desired exhaust-valve movement.

The Greene engine is thus a modernized "plugtree" valve-gear. Its valves are four in number—two steam and two exhaust—and are of the flat-slide pattern. The power which moves them is applied parallel to, and in line with, their seats. The steam-valves when tripped are shut by a combined action of a weight and the pressure of the steam on the large valve-stems, D, D', insuring a quick cut-off, and the positive closing of the port, under all circumstances of speed and pressure. The steam-valves are operated by toes, BK, $B'K'$, on the inner ends of two rock-shafts, A, A', that connect with the valve-stems outside the steam-chest. There is a sliding bar, J, carrying tappets, G, G', which receives a reciprocating rectilinear motion from an eccentric on the main shaft. Below the sliding bar is a gauge-plate, HL, connected with the governor by the rod F, which receives an up-and-down-motion from a reverse action of the governor-balls.

The tappets in the sliding bar are attached to the gauge-plate, and elevated or depressed in the bar by the action of the governor. As the sliding bar moves in the direction of the arrow, one of the tappets is brought in contact with the inner face of the toe on the rock-lever, causing it to turn on its axis, thereby opening the steam-valve at one end of the cylinder. At the same moment the other tappet comes in contact with the outer face of the other toe, and as the surfaces are bevelled, the toe is forced up into the socket until the tappet passes under, when it drops by gravity alone into its original position to be operated upon in its turn, when the motion of the sliding bar is reversed.

As a result of this motion, as the bar moves in a straight line, while the toe describes the arc of a circle, the tappet will

pass by and liberate the toe, which is brought back to its original position by a weight, and the steam-pressure on the large valve-stem, which thus closes the valve and cuts off the steam. The liberation of the toes will take place sooner or later, according to the height of the tappets; that is, the lower the tappets are, the sooner the toes will be liberated, and *vice versa*. By the elevation or depression of the gauge-plate, the period of closing the valves is changed, while the period of opening them remains the same. The adjustment of the gauge-plate is effected directly by the governor.

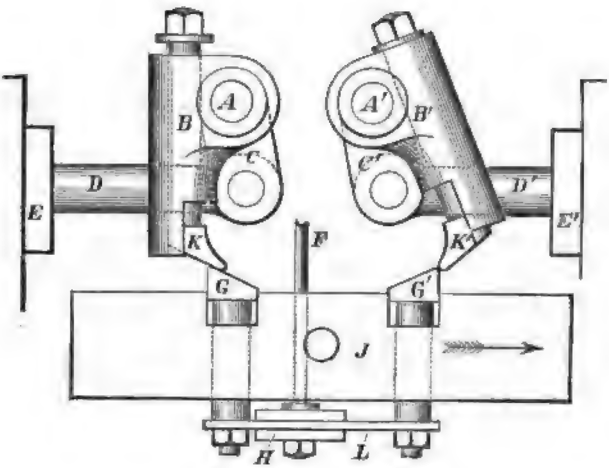

FIG. 122.—THE GREENE TRIP-MOTION.

In designing the Greene valve-gear, the independence of the steam and exhaust sides, especially where two eccentrics are employed, permits more freedom in determining the elements of its kinematic action than is possible with the ordinary Corliss gear. The exhaust system may be arranged practically without other regard to the steam side than to see that the steam cannot at any time "blow through"; i.e., the exhaust must always close before the steam-valve is opened. The exhaust-valves are of any desired type; but they should usually be of such kind as to secure prompt and full opening, and unimpeded egress, for the steam and cushion, in just the

amount required, without producing wasteful back-pressure. The steam-valves may be flat or cylindrical, plain or gridiron, or other; but they should be as nearly balanced and frictionless as possible, and, like the exhaust, should permit a minimum volume of clearance- and dead-space. The tappets and rock-shaft arms should be so proportioned and adjusted that they will engage with certainty, open the port properly, giving correct lead, and release the valve at the proper point without causing variation or oscillation of the governor. The closing apparatus, whether weight, spring, or steam-pressure, should be so powerful and so free from liability to stick, as to make it certain that the valve will invariably close promptly and without jar. The maximum cut-off should be made as nearly full-stroke as is practicable; although it is rarely of advantage to follow beyond three-fourths stroke, even to secure maximum power; the loss in back-pressure beyond that point often nearly or quite compensating the gain on the steam side. The exhaust may often be made a cam-system with advantage.

In the Greene, as in the Corliss engine, it is easy to secure great length of port transversely to the axis of the cylinder and thus a very short port parallel to that line, and, consequently, small travel of valve in opening and closing it. The Author, in designing these engines, has sometimes gone so far as to make the first dimension exceed the diameter of the cylinder, and the second as little as one tenth or one twelfth for the steam-port, and one eighth or one tenth for the exhaust-port, the speed being moderate. Their areas should, however, in all cases, be such as to keep the maximum velocity of flow of the steam down to or well within the limits prescribed by good practice.

In designing this valve-motion, as in most cases, it is advisable to lay it down on the drawing-board and to proportion it to its work by graphical construction. The motion of the valve is the same as with any other eccentric system; but its time as related to the motion of the piston is widely different. Since it commences its stroke with the piston, lead being neglected, the travel of the valve, up to cut-off, is proportional to

the versed sine of the arc swung through by both crank and eccentric, and the port-opening is measured, when there is no alteration of stroke by the introduction of the rock-shaft system, by

$$s = R - R \cos \theta - l;$$

when R and l are the eccentric-radius, and the initial lap of the valve, measured in similar terms with s; θ being the angular movement of crank and eccentric. The lead must be sufficient both to take up the lap, l, and to give the desired opening at the commencement of the stroke. As in other trip-gears, the lead and the lap both restrict the maximum range of cut-off.

Cam Valve-motions have been from a very early date employed for the purposes which are usually subserved by the eccentric. In the figure, A is the valve-stem, B the cam-shaft and C the cam, as often employed in cases in which the poppet-valve is used as a steam-valve. The cam-shaft makes its revolutions at one half the speed of the engine-shaft, and the projections, D, E, engage the roller supporting the valve-stem at the beginning of the stroke, raising the valve to its full height, holding it up until the desired point of cut-off is reached, and then dropping it to its seat again. By giving the raised portion greater or less extent, the ratio of expansion is made less or greater as the conditions sought may dictate. By placing a series of these cams side by side, with different arcs of contact, a number of ratios of expansion become available, the cluster being shifted along the shaft to bring any cam under the valve-stem as may be required, either by hand or by a connection with the governor. In the latter case the cam is sometimes made continuously tapering from the larger to the smaller end, all ratios of expansion being thus obtained, from unity to infinity. The so-called " French cam" is of this form. This device has also been incorporated into one of the types of valve-gear designed by Mr. Wright.

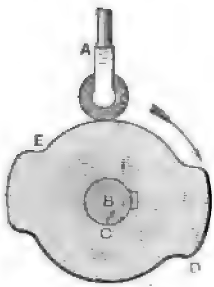

FIG. 123.—A CAM.

Various other cams are used in the older forms of engine.

The "wipers" or "toes" used in the so-called American steam-engine, the Stevens engine, in the Stevens and Sickels cut-offs, are properly classified as cams. In the earlier locomotive valve-motions, a heart-shaped cam was often employed to operate the three-ported slide-valve. Straight-sliding wedges and driving-pieces of irregular outline, which are often found to give very perfect relative motions, belong to the same class. Cams cannot be successfully used, however, in any fast-running engine. The irregularity of their action and the consequent jar and wear, or even liability to fracture of connecting parts by the inertia-stresses and strains so introduced, forbid their general adoption.

Cam-motions cannot usually be employed satisfactorily on engines of high speed of rotation, and are limited in this respect even more than the releasing-gear of the types just described. On large and slow engines, however, as on steam pumping-engines, they have been sometimes found to work well. As their changes of speed are more abrupt by far than the eccentric-gears, care must be taken to provide against serious shocks and resulting damaging blows. Where they are admissible, the facility offered by the cams for securing any desired motion gives them great attractiveness to the designing engineer. Sliding cams operated by an eccentric form a combination which has perhaps been as successful as any.

62. "Positive" Valve-gears are necessarily employed wherever the speed of engine is such that detachable gears and cam-motions become impracticable. In all examples of this class, the movements of the valves are effected through the operation of undetachable connection in kinematic combinations capable of giving the desired steam-distribution at whatever speed the machine may be operated. Such combinations are illustrated by the examples, descriptions of which follow.

Automatic Gear with Shaft-governor, as first introduced on portable engines by Hartnell, has become recently a common and a standard type for high-speed engines. All the detachable valve-gears, such as the Stevens, the Sickels, the Corliss,

and the Greene, if driven up to higher than their customary speeds, approach a limit at which the valve, when released, drops to its seat too late, and the device becomes inefficient as an expansion-gear. A speed of engine can readily be imagined at which the valve, detached early in the stroke, might not reach its seat before the crank had turned the centre. A "positive" system, as it is technically called, is thus essential with the class of engines generally adopted for dynamo-electric and other fast-running machinery, and the "automatic" engines of modern invention supply this need.

These forms of valve-gear, as already seen, are nearly all of one general type, consisting usually of a single three-ported valve driven directly by an eccentric loose on the main shaft, and adjusted either as to its lead or its throw by a governor fixed beside it on the shaft. The method of study of the action of the valve for any given position of the eccentric is precisely that for the same valve with fixed eccentric: the lead and travel are known, and the lap and dimensions of valve being determined, the distribution of steam effected by it become readily ascertainable. In actual operation, however, the governor is continually changing these relations, and the designing of the gear involves the determination of the method of variation of steam-distribution with change of speed and of position of governor. It is evident that alteration either of lead, as when the governor turns the eccentric on the shaft, or of travel, as when the governor alters the eccentricity and throw by moving the eccentric across the shaft, will change the amount of work done in such manner as to give not only a system of regulation, but also a fairly good distribution throughout any ordinary range of load and speed. The effects of both methods have been shown already in § 54, which may be again consulted. Several examples of this type are illustrated in the succeeding articles.

The problem to be solved by the designer in constructing any valve-gear of this class involves a study of its elements with a view to securing a good steam-distribution and satisfactory regulation. The essential elements will usually be

found to include balanced valves, large port-areas, comparatively short stroke and great length of port transverse to the cylinder, a shifting eccentric earefully balanced, and a governor comparatively frictionless, nearly isochronous, and in some cases controlled by a dash-pot. Such an adjustment as will give high compression with short cut-off is usually advantageous, particularly with a single valve and considerable clearance.

A disadvantage is found in the attempt to apply the shaft-governor to single-valve engines in the fact that the alternative commonly lies between varying by its action either the angle of advance of the eccentric or the eccentricity. The former necessitates serious variation of lead, increasing it with increasing expansion; while the latter system decreases lead with shortened cut-off, the two changes producing opposite lead-changes. The problem, in designing such governor-systems, is to first define the most desirable path of the shifting centre of the eccentric, and then to so arrange its moving mechanism, in its connection with the governor, as to, as nearly as practicable, trace that path across the shaft-line. This can be done by simple methods of varying eccentricity and angular advance at the same time. Decreased travel, increased angular advance, and increased expansion may in this way be made to go together; and thus a more nearly constant lead may be secured.

63. Thompson's Automatic Gear, commonly known as the "Buckeye," was one of the earliest of this class to become successfully and generally introduced. Two valves are used, a main and an expansion valve.

The live steam enters at *D*, Fig. 124, whence it passes through passages *a a''* and the open pistons *FF*, into the interior of the box slide-valve *BB*, as shown by the arrows. From the box-valve the steam is admitted to the cylinder through ports *b b* in its face, as these ports are alternately brought to coincide with the cylinder-ports. The cut-off valve, which consists of two light plates, connected by rods works on seats inside of the main valve, as shown, and alter-

nately covers the ports leading to the cylinder. The cut-off valve-stem *g* works through the hollow stem of the main valve. The exhaust takes place at the end of the valve (as shown by arrows on right), into the valve-chest and into the

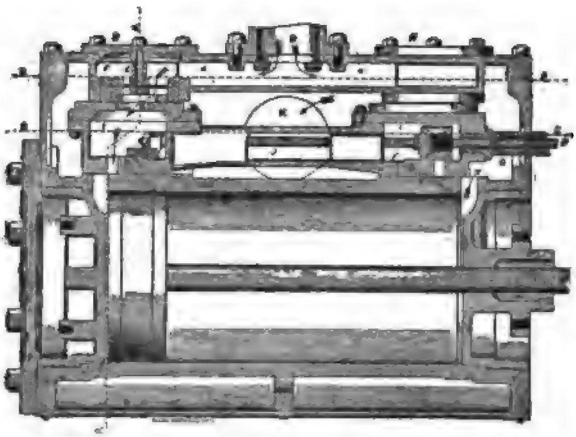

FIG. 124.—THE THOMPSON VALVE-GEAR.

exhaust-pipe below. The expansion-valve is actuated by an eccentric on the main shaft controlled as to its position by the governor, and, being turned on the shaft in such manner as to vary its angle of lead, the time of suppression is correspondingly varied, and the ratio of expansion is adjusted to the work momentarily demanded. The travel of the valves being invariable, the expansion-valve always does its work nearly at the time of maximum speed, and thus gives a sharp cut-off "corner" on the diagram and maximum extent of adiabatic expansion, and approximately uniform wear is assured.

The valve-gear is so set that the expansion-valve does not close the port when the governor-weights are at the inner extremity of their range, but holds the port continually covered, or nearly so, when driven to their outside limit by the action of centrifugal force of excessively high speeds of rotation. The springs may be so adjusted as to location, length, and strength as always to precisely balance centrifugal force in all positions of the weight at the proposed speed of the engine. The per-

fect isochronism thus attainable is not, however, always desirable, and the springs are commonly given considerable initial tension. The theory of this governor will be given elsewhere.

64. **The Armington-Sims Valve-motion** was one of the earliest of the single-valve variety to give satisfactory results in economy of steam and in regulation.

The valve is a balanced piston-valve, which gives us a very quick admission of steam and small clearance. Such valves are more liable to leak than are flat valves, but they are sometimes in use for ten years or more, and remain perfectly tight; properly made and with good feed-water and good lubrication, they will keep tight for long periods of time. These valves may be renewed at very small cost.

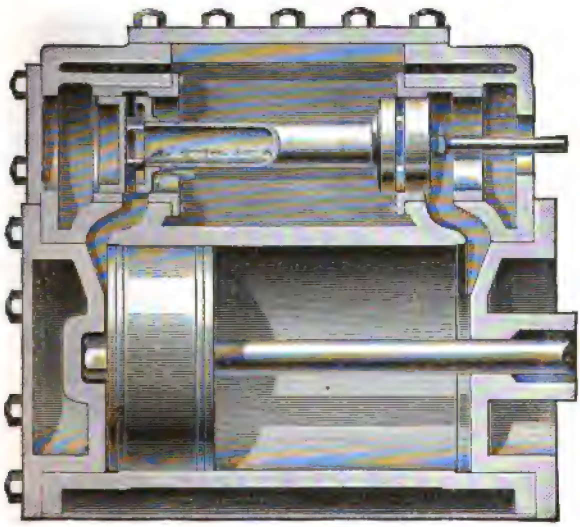

FIG. 125.—THE ARMINGTON-SIMS VALVE AND CYLINDER.

The steam-chest and valve-seat are in one casting with the cylinder; the valve-chest is enclosed by a cover in the usual manner, which enables the boring of the valve-seat to be accurately done, and gives an opportunity to easily set the valve "Live" steam here surrounds the valve, and, taking steam in the middle and exhausting at each end, the steam-ports can be made very direct, and the clearance small. In the

cut the valve is shown taking steam into the cylinder at the back end: the valve at the other end is taking steam from the steam chest *through* the valve into the same cylinder-port, giving ample port-space and a high initial pressure. Steam is exhausted at each end by direct passages, which should be so large as to permit very low back-pressure.

The piston-valve has the advantage of a perfect balance, which relieves the governor of all embarrassing load; and this form, in which the exhaust takes place at the ends, is less productive of waste by internal condensation than the more usual

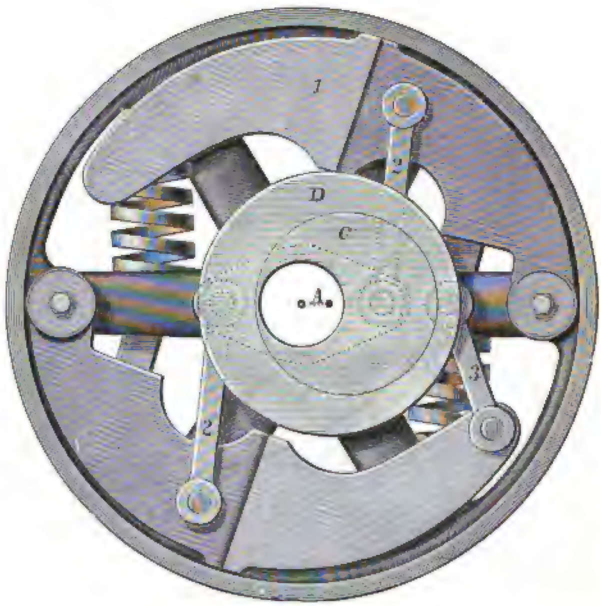

FIG. 126.—ARMINGTON AND SIMS GOVERNOR.

type. The proportions of valve- and clearance-spaces should be such as to give, at the rated load and its appropriate expansion, compression up to the steam-chest pressure, or nearly so. The governor of this engine resembles the preceding in general construction.

65. **The Sweet Valve-gear,** a system devised by Professor J. E. Sweet, illustrates still another method of varying the expansion by the action of a shaft-governor.

The valve is a rectangular block, sliding between the seat and a cover-plate; as shown in the engraving. Ports are cut through the cover-plate, through the valve, and through the seat

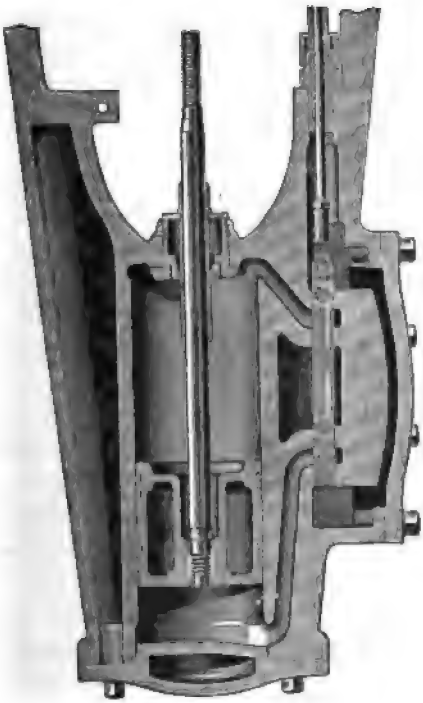

FIG. 127.—SWEET'S VALVE.

into the steam and exhaust passages in the cylinder-casting, in the proper positions. These ports are double at the ends of the valve, and a single port of ample area is made through the middle of the valve.

This valve, permitting the use of a detached cover-plate, which, while sustaining the pressure of steam that would otherwise come upon the valve, and thus making it a balanced valve, allows any unusual pressure, occurring when the piston comes back to the compression period of its cycle, to raise it, and thus to permit the water which may have caused the press-

ure to flow back, so relieving the cylinder, and obviating all danger of forcing out the heads. The valve-cover is sustained on loose packing-strips, which are free to close up upon the edges of the valve, and to take up wear as it occurs. The form of the plate is such as to give it great stiffness against the superincumbent pressure, and thus to prevent pressure on the valve itself in consequence of spring in the plate, and the ports are so placed as to prevent the cutting away of the faces and seats by the currents of steam.

The valve is driven by an eccentric, the motion of which is controlled by the governor, and the connection of which with the valve is effected by a peculiar system of linking. The eccentric is so suspended from the disk, to which it is attached, that it may be thrown across the shaft by the action of the governor, in such a manner as to give the effect of the once common "Dodd motion." It is carried on a lever, which is pivoted at one side of the shaft, while the governor-rod is attached at the opposite side. The singular positions of the eccentric-rod and the rock-shaft arm enable the alteration of the throw of the eccentric produced by the governor to be effected without alteration of the lead of the valve, so that the steam may be admitted, at all times, at the same point in the revolution of the engine. This it does, since the line of the eccentric-rod is, at the commencement of stroke, in line with the lever on which the eccentric is carried.

The governor consists of a single weight, or ball, carried on the end of a lever which is pivoted, near its middle point, on one of the arms of the governor-pulley, and connected to the spring, by which it is held under control, by a link, extending across to the other side of the shaft to the end of the spring, which is there secured to the rim of the pulley. When the speed decreases, the tension of the spring, at the end of the weight-lever, overcomes the centrifugal effort of the ball, and the latter is forced in toward the shaft, carrying with it the end of the eccentric-lever, and thus giving the valve greater throw, and extending the period through which the steam follows the piston, producing more power and bringing the en-

gine up to speed. The reverse change of speed of engine produces the opposite action of the eccentric and of valve-motion, and the cut-off is shortened, and the power of the engine is reduced to that needed to give the correct speed. As such governor may be made as nearly isochronal as may be desired, the approximation to correct speed may be made as close as is consistent with the sensitiveness considered permissible. The use of a single eccentric and of a single governor-ball, and the general simplicity of this combination, are characteristics of this design.

66. **The Allen (Porter) Gear** is one which, among the earliest of those of the automatic type, has been found to give better steam-distribution than many of later design. It is a modification of the Fink system, and was introduced originally by Messrs. Chas. T. Porter and John F. Allen in the Porter-Allen engine, the pioneer of this group of steam-engines. The earlier forms of positive gear gave a very unsatisfactory distribution of steam. They usually produced a "throttling" of the steam at the point of cut-off, which was not at all such as would satisfy the engineer, and the load which they threw upon the governor was a fatal defect, as it was then arranged and connected.

This arrangement, as already described in Chapter I, has been seen to consist of a single eccentric driving a link-motion, to operate the steam-valve and to work the exhaust at the same time. The link is controlled by a Porter governor, so connected that the gear may be adjusted by the governor to any desired cut-off.

As has been seen, this gear is operated by a single eccentric, its connections so arranged that both steam- and exhaust-valves may be worked from it and any desired variation of the expansion-ratio given by the governor. The eccentric is set with the crank, and the bell-crank connection and the link which form a part of it enable all required motions to be obtained. Crank and eccentric are sometimes both forged with the shaft. The accompanying diagram by Mr. Porter will

show how these movements may be traced and their effect on steam-distribution determined.

Here, *O* is the centre of the shaft; *AB* is the path of the crank; *CHDI* is the path of the centre of the eccentric; and *EKF* is the arc in which the trunnions of the link vibrate, about *G*, the sustaining pin.

This arc is divided into twelve parts, the terminations of which are indicated by numbers from one to twenty-four. The same numbers indicate corresponding points in the path of the centre of the eccentric. The curved lines show the positions of the centre line of the link, corresponding with these positions of the eccentric and trunnions. *KH* and *KI* represent the line connecting the centre of the trunnions with the centre of the eccentric, at each extreme of its vibration.

When the engine is on either dead-centre, the centre of the trunnions of the link stands at *E* or *F*; the line connecting this centre with the centre of the eccentric coincides with the line of centres *DF*; the centre line of the link stands on *EM* or *FL*, and the block can be moved from end to end of the slot without imparting motion to the valves. The engine has constant lead for all points of cut-off.

The horizontal throw moves the link from one to the other of the lead lines, *EM* and *FL*; which motion draws off the lap of the valves. The opening movement is produced by the tipping of the link alternately in the opposite directions beyond the lead lines, and these motions are given by the vertical throw of the eccentric.

The angular vibration of the connecting-rod causes a considerable difference in the motion of the piston in the opposite ends of the cylinder, and when the length of the connecting-rod equals six cranks this difference in velocity averages 20 per cent, and at the commencement and termination of the strokes reaches 40 per cent. The driven arm of the link is proportioned to the eccentric-throw as the length of connecting-rod is to the crank-arm; its angular vibrations, to *KH* and *KI*, coincide, in degree as well as in time, with those of the connect-

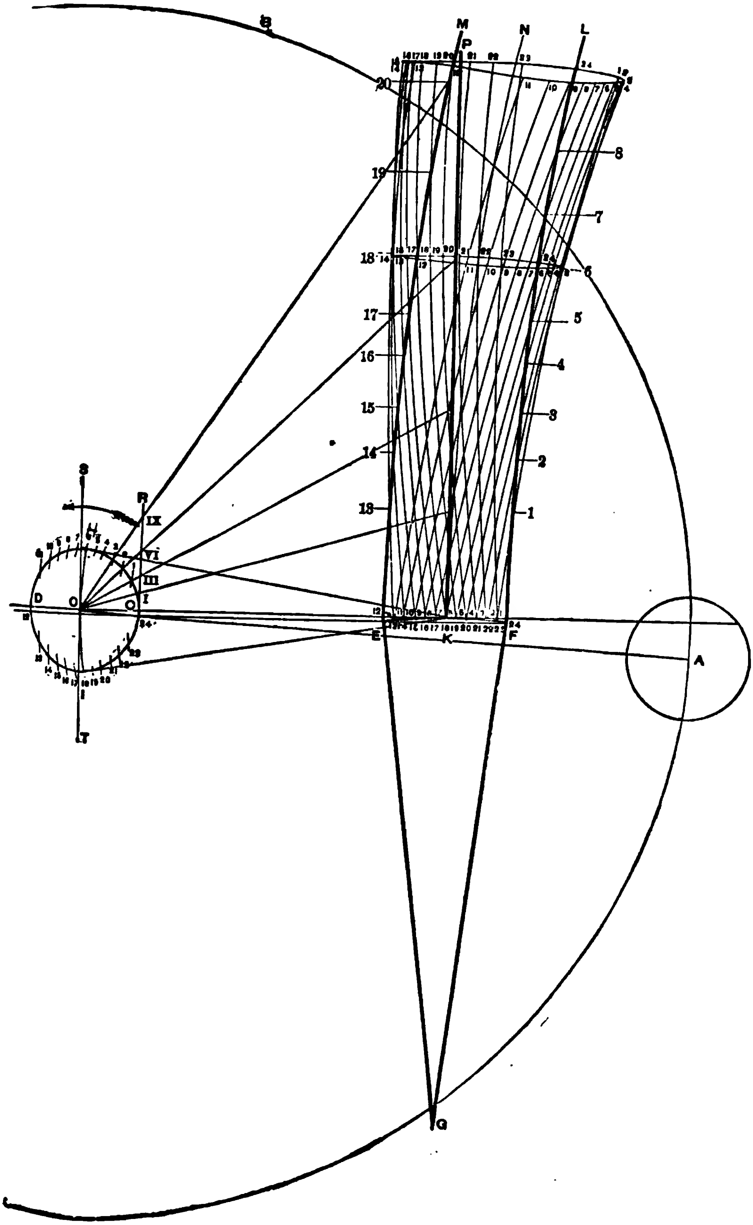

FIG. 128.—THE ALLEN MOTION.

ing-rod. The link gives to the valves different velocities corresponding to the difference in the velocity of the piston.

The tipping of the link in the direction *from* the shaft is produced by the upward movement of *C*. If *F* were at rest, *C* would describe an arc about *F* as a centre. But *C* is compelled to move in the arc *CH*, and so draws after it the point *F*, through a distance equal to the interval between these diverging arcs, limiting the opening and causing an early closing of the valve. The tipping of the link *towards* the shaft is produced by the downward movement of *D* at the extremity of *DE*. If *E* were at rest, *D* would describe an arc about *E* as a centre, and this arc would be drawn with a longer radius than *DI*. *D*, moving in the arc *DI*, draws after it, therefore, *E* more slowly, permitting a wider opening and a later closing of the port.

The action of the link would be mathematically more correct if the trunnions, instead of vibrating in an arc, moved in a direct line, *EF*, along the line of centres. The lead would be constant, and at mid-stroke the opening and cut-off on the opposite strokes would be equal. If, with radius once and a half the driven arm of the link, an arc be drawn below the line of centres, so that this line is tangent at *O*, ordinates drawn to this arc, from points in the path of the eccentric corresponding to equal advances of the piston on its opposite strokes, are practically equal in length.

Let, therefore, the sustaining pin *G* be lowered the short distance necessary to cause the trunnions to vibrate in the arc shown, tangent to the line of centres at *K*, and it will produce equal valve-action on the opposite strokes at every point of cut-off.

The link will now arrive on its lead-line *FL*, when the crank is at *A*, before it reaches the line of centres for its forward-stroke, while, on the return-stroke, it will not arrive on its lead-line *EM* until the crank has reached a position corresponding to *A*, after passing the line of centres ; the driven arm of the link standing then on a line below and parallel with the line of centres, as shown.

This adjustment gives a difference of lead in favor of the centre where the motion of the piston is most rapid; a valuable feature, since it is necessary not only that the motion of the valve shall be accelerated, but also that the area of opening shall be proportionately enlarged, and this enlarged area is furnished by the greater lead.

The exhaust-valves open and close their ports as the centre line of the link crosses the line *KN*.

The movements of the link are the same that would be imparted by a series of eccentrics of increasing throw and diminishing angular advance. To show this:

At the point *K* erect *KP*, and through *O* draw the perpendicular *ST*, and *CR* tangent to the arc *CH*. Then, from *O* draw a line, terminating at any point on the perpendicular *KP*. A portion of such diagonal will form a secant to the tangent *CR*. Four illustrations of these diagonals are given, forming the secants *O I*, *O III*, *O VI*, and *O IX*.

The identity of the link with the eccentric is shown thus:

(1) The length of the secant is equal to one half the throw of the link, at the point at which the diagonal terminates.

(2) The section of the secant beyond the circle represents the opening movement of the link at that point.

(3) The movement of that point and the opening are therefore that derived from an eccentric having a throw equal to twice the secant, and advance equal to the angle made by the secant with the perpendicular *ST*.

(4) The intersection of the secant with the circle shows the point in the revolution at which the full opening is given.

(5) This point bisects the arc included between the commencement of the opening and the point of cut-off. The diagonal thus represents the line of centres of such large eccentric.

67. Continuously-rotating Valves are occasionally used; although none has yet become well known. The earliest of this class, so far as known to the Author, was that of Goodrum, an American inventor, which was proposed about 1855.* It

* U. S. Patent Office Report, 1855, vol. I., p. 580; vol. II. p. 127.

consisted of a cylindrical shell into which steam entered at one end, and from which it escaped laterally through longitudinal slits made to coincide with the ports of the cylinder at proper points in the revolution. It was continuously revolved in its casing at the same rate as the crank-shaft, and the time allowed for entrance of steam into the cylinder was determined by the extent of the opening on the circumference of the valve. The exhaust-valve was open to the cylinder during one half the revolution, closing the port during the other half. An adjustable lap on the steam-valve, which might be actuated by the governor, permitted variation of the point of cut-off, the ratio of expansion, and the power on speed of the engine.

An ingenious and effective modification of this device is that of Ehrhardt,* who has also adapted it to the compound engine. The Ehrhardt valve revolves at one half the speed of the engine, the operation of opening and closing each port thus taking place within an arc of 90.° One valve is employed for both cylinders of the compound engine. It is not always practicable to secure as large ports with this device as with the more common forms of valve; but its simplicity and the smoothness of its operation at all speeds fit it exceptionally well for use in engines of very high speed of rotation. It is absolutely free from all ill effects of inertia or reversed motion.

Valves of this character have been made by a number of American inventors; but they have more commonly been of disk-form, revolving about the axis of the circular disk, the ports sometimes radial but often normal to the face of the disk. The valve of Ehrhardt above described was coned slightly to permit tightening by giving it a small movement endwise. This class of valve rarely affords a sufficient area of port.

Rotating valves and rotary valve-gears, although certainly

* See Die Steurungen der Dampfmaschinen, von Emil Blaha, Berlin, 1885, S. 181–3.

known as early as the time of Thomas Goodrum, previous to 1860, have not come into common use; yet their smoothness of motion at the highest speeds of rotation, their perfect balance, their freedom from objectionable inertia-effects, and the possibility of securing good steam-distribution with good regulation would seem to be advantages that should lead to their more general use. The valve of Goodrum and the later valves of Radinger and Ehrhardt have shown the practicability of their employment, and even possibly in place of the now customary systems of variable expansion-gears.

68. A Reversing-gear is essential to the proper operation of the steam-engine in many cases, as invariably on the locomotive and with marine engines. Stationary engines seldom require them; although many rolling-mill and other engines need to be provided with means of turning backward occasionally by hand at least. There are various types of reversing-gear, most usually involving some method of either shifting an eccentric from the position suitable for going ahead to that proper for the reverse motion, or the transfer of the valve-connections from one to another eccentric.

As illustrating the first of these two methods may be mentioned the "loose eccentric," which is loose on the shaft and is driven by contact with a "lug" or "stop" on a ring keyed fast to the shaft. When going ahead, the driving lug holds the eccentric in such a position as to give it the desired lead. To reverse, the engine is stopped, and then is reversed by hand. The eccentric stands on the shaft until, the backing stop coming around and in contact with its appropriate lug on the eccentric, it is driven by the engine-shaft in that relative position which gives the lead desired while moving backward. The angular change of position on the shaft is thus 180° minus twice the angle of lead, if lead is the same for both motions. The valve-connections are precisely the same as with the single fixed eccentric. In some cases, for heavy marine engines, a hand or a steam reversing-apparatus is employed which drives the eccentric ahead into the position appropriate for the re-

versed motion. Another system provides a means of shifting the eccentric across the shaft, as already seen in the attachment of the shaft-governor for regulation, either by carrying it on a lever spanning the shaft or by means of a system of wedges, in either case producing precisely the same final change as by the method just described, the difference being simply that the change from one position of the eccentric-centre to another is effected by carrying it directly across, instead of turning it around on the main shaft. Loose eccentrics should be carefully counterbalanced.

The second class of reversing-arrangements, by which one or the other of two eccentrics can be used at will, is most commonly illustrated by the several forms of "link-motion," some of which will be presently described. In all systems of this general character, one eccentric is used when going ahead and another when reversed, and the valve-connections are shifted from the one to the other as is found desirable. In the earliest of these reversing-gears, this connection was effected by means of a hook on the end of each eccentric-rod; one being thrown out and the other then thrown into gear when the engine was reversed.

Link-motions are substitutes for the hook-motion, the transfer from one eccentric-connection to the other being effected by shifting the "block" or "slider" to which the valve-stem is attached along a slot in a link one end of which is secured by a pin to the extremity of one eccentric-rod and the other end to the other rod. Sometimes the link is stationary and the end of the valve-connection movable; oftener, the block is fixed and the link is movable by means of the "reversing-lever," which lever is placed where it can be conveniently handled by the attendant. Occasionally a "half-link" is used to shift the eccentric and valve connection in such manner as to vary the valve-motion but not to reverse the engine.

69. **The Stephenson Link-motion*** consists of a system including a forward and a backing eccentric secured in proper

* Invented by Howe in 1843 and first applied to the Stephenson locomotives.

position on the shaft, permanently connected by means of its rod to the opposite ends of a curved slotted bar called the link, as is seen in the figure, where *a* and *b* are the eccentric-rods; *cg* is the link; *h* the valve-stem; *i* the connection, through the bell-crank *ik*, to the reversing handle and screw,

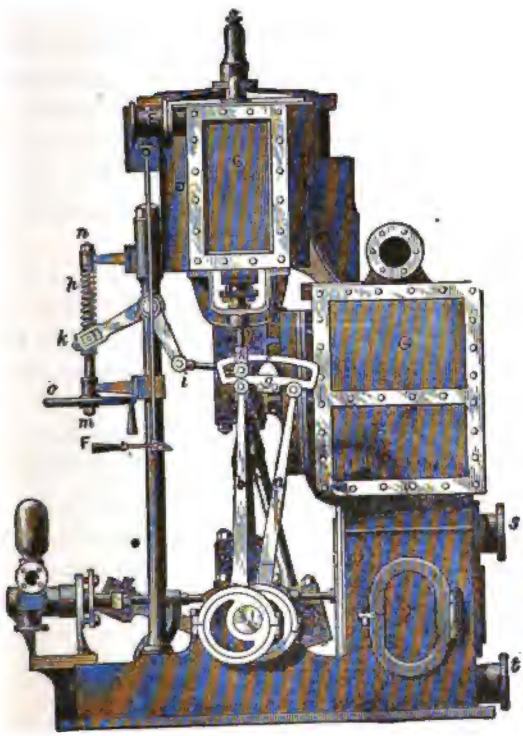

FIG. 129.—COMPOUND ENGINE AND LINK-MOTION.

mn, *op*. The illustration exhibits the usual method of application to the marine engine. The same arrangement, substantially, is adopted for the locomotive, except that the "reversing-lever" is substituted for the screw.

This is the most common of all the link-motions.

The link is said to be in full forward or full backward gear

when the link-block or slider is at one extremity of the link;

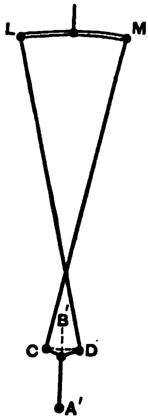

FIG. 130. CROSSED RODS.

and the motion of the valve is then that due to the action of one eccentric only. When the crank is "on the dead-centre," or "on the centre," it is in line with the connecting-rod. The eccentric-rods are said to be *crossed rods* when, the crank, *A*, being on the centre opposite the link, *LM*, the two rods, *CL*, *DM*, cross each other; if they do not cross they are said to be *open*. With crossed rods the steam-lead diminishes as mid-gear is approached; with open rods the lead increases. The radius of curvature of the link is usually exactly or approximately that measuring its distance from the shaft-centre.

In the operation of the link, it is commonly worked in "mid-gear," the one eccentric giving it the principal movement; but its action being modified by the opposite action of the other, thus giving a reduced travel and an altered lead as shown in an earlier paragraph of this article, and thus effecting an earlier cut-off as the working point approaches the middle of the link, at which point the motion of the valve is that due the lap and the lead of both eccentrics working together. Thus the link serves an excellent purpose, often, in giving a variable expansion, as well as in reversing.

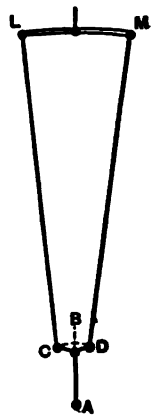

FIG. 131. OPEN RODS.

The movement of the valve is determined mainly by the throw of the eccentrics, their position on the shaft, and the location of the link-block or slider in the link, and, in a minor degree, by the method of connection of the link with the eccentrics and its suspending rod. The major conditions are fixed by the demanded steam-distribution, the minor, commonly, by considerations of convenience. This link should usually, if practicable, be suspended by a pin at or near its centre; but if habitually worked at any given ratio of expansion, the point of suspension is best fixed opposite the

position of the link-block in the link for that case, in order to avoid wear by the continual motion of the block in the link. The connection of the eccentric-rod with the link should be so made that when in full gear the rod should fall into line with the centre of the sliding block and its connection.

On inspection of the figure, which is an outline representation of the common Stephenson link, it is evident that the action of the link, AB, is determined by the position and eccentricity, oe, oe', of the driving eccentrics, the length of their rods, FF', the shape of the link, AB, the method of

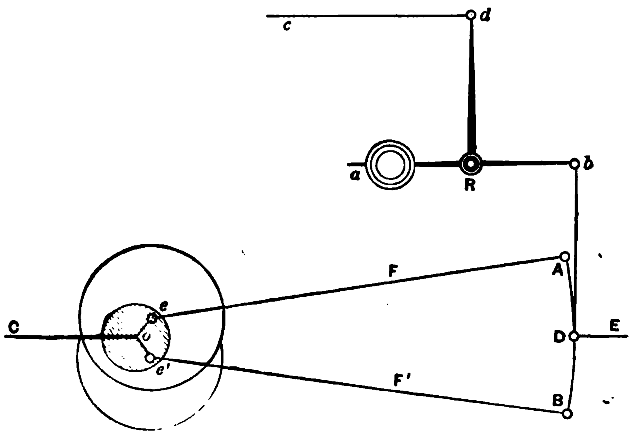

FIG. 132.—THE STEPHENSON LINK.

hanging it—in this case the system being a bell-crank and tumbling-shaft *abd*, operated by a reach-rod, *cd*, and largely also by the length of the suspension-rod *CD*—and its point of attachment. The latter should be made as long as possible, and should usually be attached at the middle, *D*, of the link as shown. The longer this rod, the steadier the action of the link. The best radius of curvature of the link is the distance to a point midway between the point of crossing of crossed links and the point of intersection of the lines of the rods, produced beyond the shaft, when open. The principal desideratum in arranging the parts connecting the link through *cd*, with the reversing-lever, is to secure a good balance and ease of operation. Constructing the link as shown in the last figure,

its motion is readily traced by the method shown in the accompanying diagram. The radii o 1, o 2, o 3, etc., etc., are successive positions of the eccentric driving the upper, or "go-ahead," end of the link, and the radii o I, o I I, o I I I, etc., etc.,

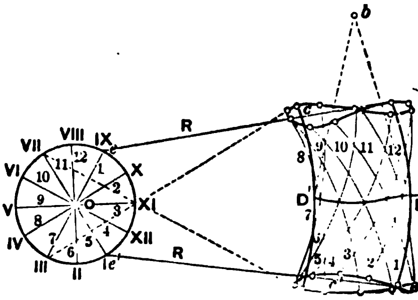

FIG. 133.—MOTION OF THE LINK.

similarly exhibit such positions for the other end. As the crank-shaft revolves, the lengths of the rods, *RR*, being constant, the simultaneous positions of the link are indicated by the lines similarly numbered on the right.

Every change of length or point of attachment of the suspension-rod, *CD*, produces a change in the forms of the curves described by the ends of the link.

For the Stephenson link in mid-gear, if we take v_1 as the velocity of motion of the valve in full forward-gear, v_2 its velocity in full-gear, running backward, and v that at any intermediate position of the link; and if d_1 and d_2 be the distances of the pin in the link-block from the forward and the backward ends of the link, respectively: then, taking the equal angular velocities of the upper and intermediate points about the lower end, we have, approximately,

$$\frac{v_1 + v_2}{d_1 + d_2} = \frac{v + v_2}{d_2}; \quad v_1 d_2 + v_2 d_2 = v d_1 + v d_2 + v_2 d_1 + v_2 d_2$$

and

$$v = \frac{v_1 d_2 - v_2 d_1}{d_1 + d_2}.$$

The exact motions can be best determined graphically. An approximation, as given by Rankine,* to the motions of the valve when the pin is in any given position is as follows: Let O represent the centre of the shaft, $\overline{OF}$ the forward eccentric-radius, $\overline{OB}$ the backward eccentric-radius, and LO a line parallel to FB. In full forward-gear, the half-throw is $\overline{OF}$, and the angle of lead LOF. When the pin is at any intermediate position on the link, join FB, in which take the point S, dividing that line in the same proportion in which the pin divides the link; then the motion of the valve will be *nearly* that corresponding to an eccentric-radius $\overline{OS}$; so that the distribution of the steam will be nearly that corresponding to the constant lap, with the diminished half-throw $\overline{OS}$, and the increased angle of lead $\angle LOS$.

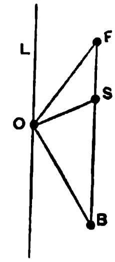

FIG. 134. LINK-MOTION.

In mid-gear, when the stud is midway between the ends of the link, the lead is 90°, and the half-throw is nearly $= \overline{OF} \times \sin \angle LOF$, being the perpendicular distance from O to FB. The nearer the link-motion is brought to mid-gear, the earlier is the steam cut off.

A more exact determination of the radius and angular lead of the single equivalent eccentric that may replace the link at any given position of the link-block is the following:

In the figure, assuming *open rods* are to be considered, let AB represent the stroke of piston and the diameter of the crank-pin circle, OC and OD the positions of the forward and the backing eccentrics, respectively, when the crank is on the centre A, and the arrow to indicate the direction of rotation of the system. Determine the point c on AB equidistant with C and with D from the two end-pins of the link and describe the circular arc CcD. Then the radius

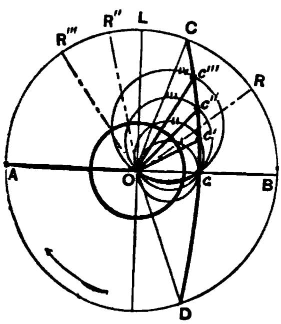

FIG. 135.—OPEN RODS.

* Steam-engine; p. 498.

Oc is that of an eccentric which would produce the same motion as the link when the latter is in its middle position and *Oc′*, *Oc″*, *Oc‴* will be radii of eccentrics operating similarly with the link-block in positions proportionally distant from the centre or the extremities of the link. Finally, drawing the several valve-circles, I, II, III, IV, as in Zeuner diagrams, for the several equivalent eccentrics, the valve-motion may be determined for each or for either precisely as with the three-ported valve and its single eccentric, the successive positions of the crank being taken as at *R*, *R′*, *R″*, for the marked points in the distribution of steam.

Where crossed rods are employed the same construction is applicable; but the arc *CcD* becomes convex toward the centre of the crank and valve-circle, and a corresponding difference in steam-distribution is observed. It will be found that the circular arc, *CcD*, is sensibly correct; but the precise curvature is parabolic.

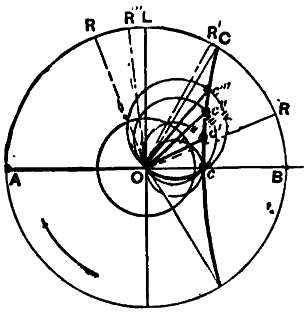

FIG. 136.—CROSSED RODS.

The location and throw of the *virtual* forward or backward eccentric being determined as above for any assumed position of the link, the application of Zeuner's or other convenient method will determine every desired element in any problem required to be solved, whether in designing or in readjusting any gear already constructed.* Examining the last two diagrams, or studying the operation of the link itself, it is seen that the effect of raising and lowering the link, and of thus altering the position of its block, is to at once vary the throw and the lead in such manner as to produce a change in the steam-distribution such that unincreased ratio of expansion is always accompanied by an increased ratio of compression.

* For a study of such problems in great detail see Zeuner on Valve-gears, or Welch on Designing Valve-gearing.

Equalized Lead, with the Stephenson link, may be secured by shifting the eccentrics slightly. As ordinarily constructed and symmetrically set, the lead varies continually as the link is moved. This is corrected, in the case of crossed rods, by shifting the eccentrics backward through an angle equal to that subtended by the half-link. With open rods, this can only improve the action in one direction, impairing the symmetry of distribution for the opposite movement of the engine. This correction is only applied, therefore, in such cases, for the forward, or most usual, direction of motion.

In designing this system, care must be taken to so locate all its members that no interference of eccentric-rod or link with the tumbling-shaft which carries the suspending-link can occur, and so that it shall be well counterbalanced and easy of reversal.

This valve-gear is not wholly satisfactory. It is subject to considerable friction, to a loss of several, often many, pounds between boiler and initial pressure, the exhaust opens and closes too early at high ratios of expansion, and neither the steam- nor the back-pressure line is of uniform height as demanded by the principle of Carnot.

70. **Gooch's Link,** invented almost simultaneously with that of Howe, is similar in its general construction to the Stephenson gear; but the link is permanently fixed in position, rotating about a pin suspended from a point in the engine-frame, and the link-block, or slider, is shifted, the valve-stem being connected with it by means of a radius-bar which is swung up and down in the link by the reversing-lever and its connections. This link is concave toward the valve-rod. The process of solution of the various problems arising in connection with it is similar to those employed with the preceding type, and the virtual eccentric is determined in the same manner. It will be found, however, that the curve *CcD* in the preceding figures is transformed into a straight line.

The next figure exhibits a common arrangement of the Gooch link. The radius, *HE*, of the link, *AB*, is the length of the valve-rod. Other parts are substantially as in the Stephen-

son system. The link is carried, however, on a fixed point, *H*, at a distance, *HO*, from the shaft-centre equal to the length of eccentric-rods. With this arrangement it is evident that the valve-rod may take its motion from any point on the link and yet no alteration of lead or of angular advance will take place.

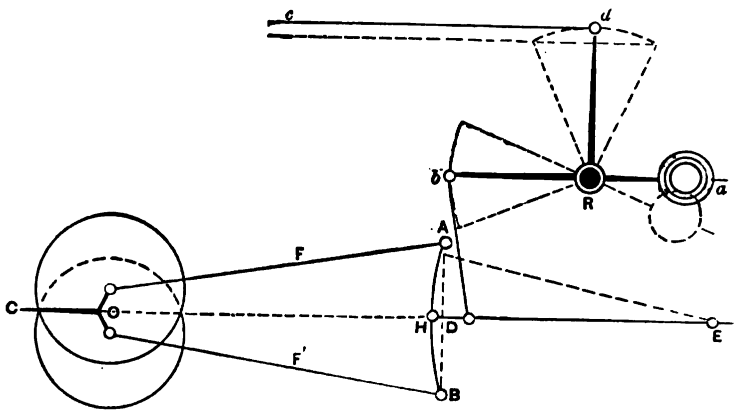

FIG. 137.—THE GOOCH LINK.

Allan's Link is so designed and constructed that both link and valve-rod are shifted, and in opposite directions, producing an arrangement and a steam-distribution intermediate in its action between the Stephenson and the Gooch systems. Neither of these forms is as commonly used as that of Stephenson ; although each possesses its own special advantages. The Allan arrangement is to a certain extent self-balanced and thus evades the necessity of introducing a balancing system, as in the case of the other links. The lead, with the Gooch, or stationary, link, is not dependent on the arrangement of the eccentric-rods, as in the case of the Stephenson or movable link, and in less degree the Allan.

71. Walschaert's and Strong's Gears are modern variations of the earlier forms of link-motion in which it is sought to dispense with their eccentrics, parts of the system which, especially in fast-running engines, are always liable to give trouble, and which are also productive of sensible frictional

losses. The Walschaert valve-gear is a kinematic chain in which the motion of the cross-head is introduced to assist in giving the desired movement to the valve as seen in the accompanying illustration, a single eccentric, or supplementary return-crank as here shown, being employed to give the valve its principal motion; but it is set, without angular advance, at right angles with the main crank, thus, by the combination of the two movements, securing a movement of the valve similar to that obtained with the Gooch link and its two eccentrics. In some locomotives the motion given by this single eccentric has been taken from the cross-head of the opposite engine, and in still other systems, from the main connecting-rod, or other point, as in the systems of Brown, Marshall, and Joy.

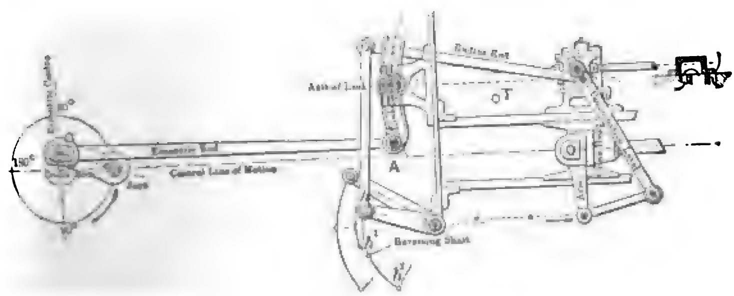

FIG. 138.—THE WALSCHAERT VALVE-GEAR.

In the Walschaert gear, the link is suspended from a fixed pin; one end being driven directly by the eccentric-rod and and the link vibrating about the pin. The valve-rod is steadied by a cross-head, and actuated by a vibrating link which is attached indirectly to this cross-head at one end, and to the link-block, or slider, at the other, the position of the block and the motion of the valve and of the engine being determined by a reversing-lever handled by the engine-driver as in the usual case. A lever hung from the cross-head guiding the valve-stem has its lower end linked to the main cross-head and furnishes a point for attachment of the vibrating, or radius, rod, in such manner as to give the required advance.

Reversal is effected by shifting the link end of the radius-bar from one side to the other of the link-centre, and the valve may be given constant lap and constant lead.

72. **Strong's Valve-gear** is a modification of the system in which the Walschaert gear is classed, and has been successfully employed on locomotives, its designer endeavoring to secure those essentials to satisfactory working at high speed: free admission; widely variable expansion; prompt opening and late closing of the exhaust; compression to boiler-pressure; and ease of adjustment and reversal. The gridiron valve is used, with a kinematic system of movement which, like that of Corliss, permits little motion under pressure, but quick and full movement while opening or closing, thus securing, at even high speeds, approximately, boiler-pressure in the cylinder.

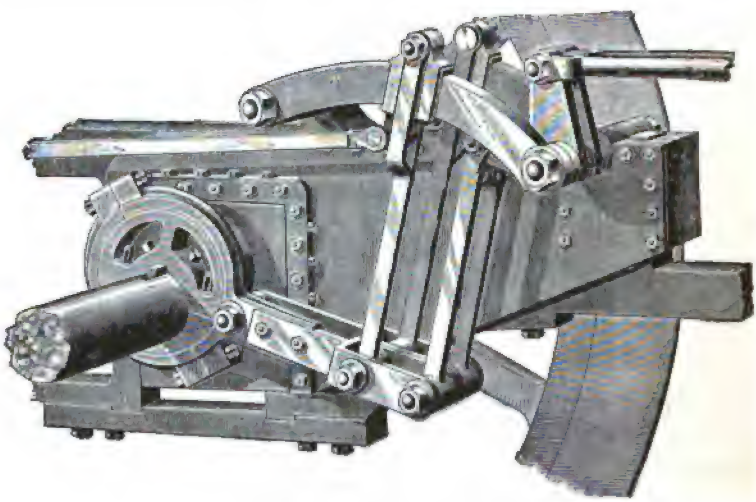

FIG. 139.—STRONG'S VALVE-GEAR.

The motion for the valves is obtained from a single eccentric, one motion of the lever attached to the eccentric moving the valves by the amount of their lap and lead, and another producing the opening. There are two levers worked from the same eccentric-strap; one being bolted rigidly to it, while the other has a pin forged on its end. This pin has a bearing

in a bushed hole in the strap. Both levers have a fulcrum-pin, connected with one end of a link, whose other end is hung by means of a pin from a block, capable of being moved along a sector or arc. The path of the pin when moved along this arc is radial to the fulcrum-pin already mentioned. Thus the position of this block on its sector, which is regulated by the lever in the cab, determines the inclination of the travel of the fulcrum-pin. When the block stands in the centre of the sector, as shown on the drawing, the valve is moved only the amount of lap and lead. If the block is moved forward on the sector, the fulcrum-pin travels over an inclined path, which determines the opening of the valve in addition to the lead, and the engine moves forward, and if the block is moved forward to the end of the sector, the full travel of the valve is given, and steam follows the piston nearly full stroke; if, on the other hand, the block is moved back past the centre, the path of the fulcrum-pin and the motion of the engine is reversed. Thus it will be clearly seen that, by varying the position of the block on the sector, the travel of the valve is varied as well as the point of cut-off. In all cases the exhaust-valve is allowed to travel its full stroke, and as it is worked from a separate fulcrum-pin, with an independent link, block, and sector, its travel may be varied at will, as may be that of the steam-valve. In ordinary working the exhaust-block is never moved on its sector, except for reversing, when both steam- and exhaust-blocks are moved at the same time. This gear has been made to give an initial pressure very nearly equal to boiler-pressure, cutting off at one-sixth stroke, opening the exhaust quite at the end of the stroke, and compressing from three to four inches on a twenty-four-inch stroke.*

73. **Brown's and Hackworth's Valve-gears** are successful attempts to dispense with double eccentrics by another system. The eccentric is keyed on the main shaft directly opposite the crank. The eccentric-rod is supported at its outer end by a sliding block, of which the path may be given greater

* Journal Franklin Inst., Feb. 1888, p. 93.

or less inclination, and on either side the line of the rod, so that one end of the line of the rod describes a circle concentric with the shaft, while the other moves in a straight line parallel or inclined to that of the rod. The guide-bar being set at a greater or less inclination, the travel of the valve is varied, and its inclination being reversed with respect to the line of the rod, the motion of the engine is reversed. The valve-rod is attached to the eccentric-rod between the eccentric and the sliding block, and the point of attachment moves in an elliptical orbit, the axis of the ellipse having directions and sizes, relative and absolute, which are determined by the proportions of the mechanism and the position of the guide-bar.

This device permits dispensing with an eccentric, gives a better motion of valve than the link-motions, especially at short cut-off, and enables the designer conveniently to locate the valves and chests on the front of the cylinders, and thus to place the engines closer together.

Marshall's arrangement guides the head of the rod in the arc of a circle by means of a radius-bar instead of a guide-bar, and is thus less subject to friction.

Angstrom's modification employs a pair of radius-bars as the guiding system, and secures a still better result* by thus obtaining a straight path for the head of the rod by the use of a parallel motion. A constant lead and symmetrical motions for both ends of the steam-cylinder are obtained.

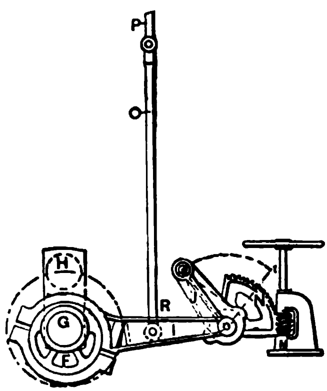

FIG. 140.—THE MARSHALL GEAR.

The Marshall valve-gear, known in Germany as that of Klug, is shown in the figure as fitted to marine engines. It consists of the eccentric *F* fixed on the crank-shaft opposite the crank *H*, the eccentric-rod *I* having its outer end

* Trans. Am. Soc. Mech. Engrs., Vol. v.; 1884.

attached to a radius-link *J*, which vibrates on a pin at the end of a lever *K*, carried by the shaft *L*. At an intermediate point of the eccentric-rod is attached the valve-rod *O*, which is coupled to the valve-spindle *P*. The slide-valve is double-ported at its upper end, but at the lower end gives a single opening only, the whole area being the same top and bottom.

The action of the Marshall gear can be compared with the system of operation by eccentrics or by link by laying down the curves of motion of valve and piston, as in the next figure. It is seen that the former has been given more symmetry of movement than, for example, the Allen with which it is here compared, or the common motion seen in Art. 54.

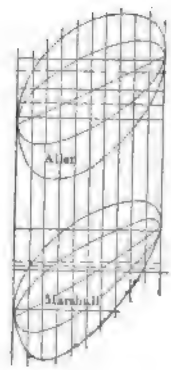

FIG. 141.—ALLEN AND MARSHALL VALVE-MOTIONS.

74. **Joy's Valve-motion** is a further modification which dispenses with eccentrics entirely. Its application to the marine engine is illustrated in the accompanying engraving. The motion is derived wholly from a point, *A*, on the connecting-rod, a vibrating lever, *AB*, swinging about the lower end of a bar, *BC*, pendent from the engine-frame, giving precisely the same initial system of movements seen in its various applications in the Brown, the Hackworth, and other so-called "radial valve-gears. The rod, *DE*, taking its elliptical motion from the swinging bar, *AB*, and guided longitudinally and laterally at the other end, *E*, by a curved bar or link, *FG*, actuates the valve-stem, *H*, through the valve-rod, *EH*, the pin at the lower end thus receiving a motion giving lead, by the vertical movement of the main rod, *A*, while the principal travel is produced by the lateral movement along the guide-link, *FG*. The latter is pivoted on the frame, and its inclination being varied by a hand-wheel or a reversing lever, as at *I*, the variation of the point of cut-off and the reversal of the engine are effected as with the forms of radial gear already described.

The same range of application is secured here as in the other valve-gears of this type, and it is perfectly possible, with

careful designing, to obtain quick and equal expansion both sides the piston, equal lead, and a good exhaust. The cost of this construction is stated to be a third less than that of the older Stephenson gear, and its action better. Frequently a swing-link carries the point *E*, instead of the guide *FG*, and, the same path being thus given *E*, the same action of the valve

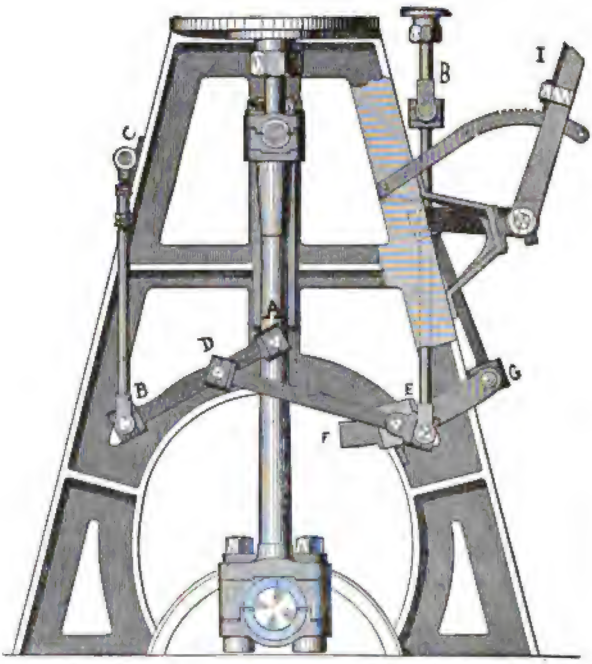

FIG. 142.—JOY'S VALVE-MOTION.

is secured. This has the advantage of freedom from sliding friction.

The following is a method of graphical construction of the Joy system, for determining the position of its guide, *G*, and the port-opening:

With the valve so placed that no lead is given, and the engine assumed to be "on the centre," draw the arc *cef*, the "lap-circle," with a radius equal to the length of the valve-rod *D*. When this rod sweeps through this arc, no motion will be

given the valve. Whenever, also, the end, e, of the link is on this circle, the port will be closed; hence, at cut-off, e must always lie on the lap-circle.

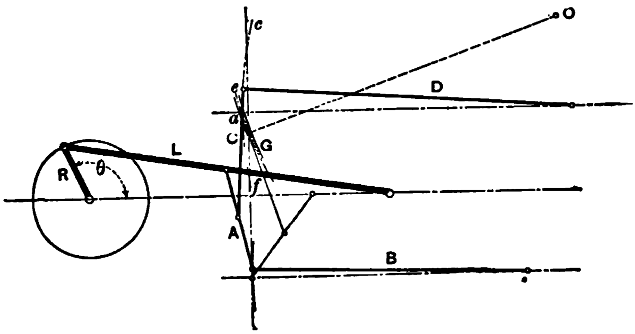

FIG. 143.—THE JOY GEAR.

To find the angular position and motion of the guide, G, draw the connecting-rod and links for the position at which cut-off is to occur, and the valve-rod D, with its extremity at e.

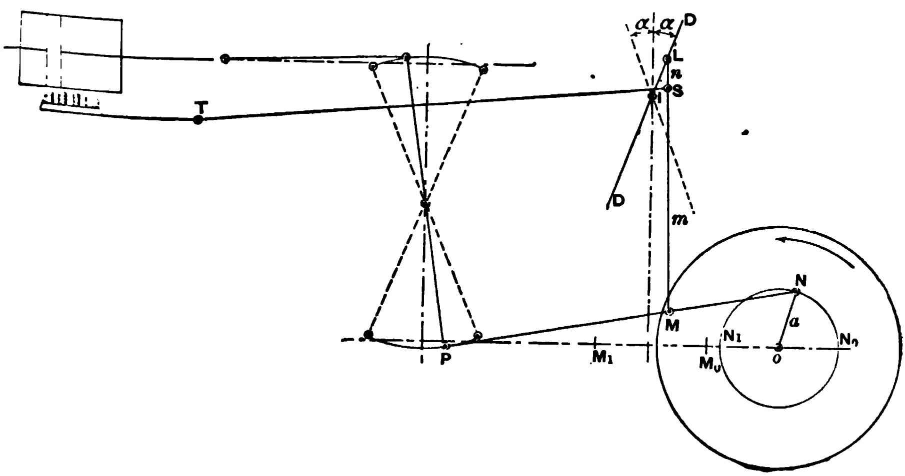

FIG. 144.—WALSCHAERT'S VALVE-GEAR.

Then C being the centre about which G is to move, and e a point in its arc, the guide-block falls at a, and the position of the guide is obtained, and its arc is struck, at that angle, from the centre O. Constructing the now completed system in any

other of its successive positions, the valve-opening will be measured by the departure of e from the lap-circle cef.

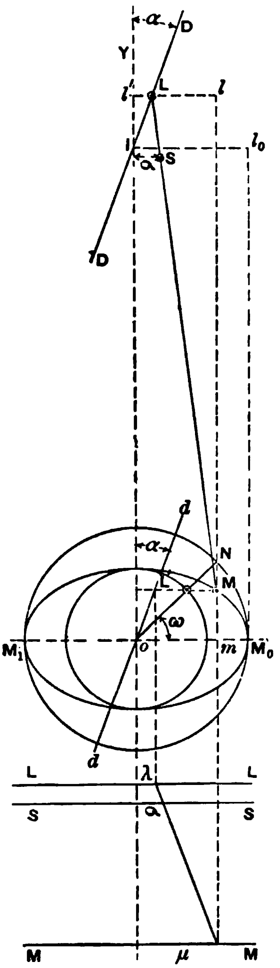

FIG. 145.—WALSCHAERT'S GEAR.

Steam-distribution without eccentrics has long been practised, and, in some instances, with success. Of this, the Walschaert gear was probably the first example of such a complete kinematic chain in which lead was secured. There are now many such in use. The following simple treatment will illustrate the geometry of this class: *

In Fig. 144, the diagram represents a Belgium tramway-engine gear. A link, ML, connects a point, M, on the connecting-rod, with a slide, L, on an inclined director, DD, which is rectilinear. With the link, a bar, ST, connected with the valve-stem at T, gives the latter the motion due the horizontal movement of the point S.

Let $ON = a =$ the crank-radius;

$Pm \div Pn = k$;

$\omega =$ the angle of rotation, N_0ON;

$SL = n$; $MS = m$;

$a =$ the angle of DD with the vertical.

The point M will move in an ellipse; and its projections on NP will be approximately obtained from the projection, at the instant, of N on N_0N_1. The location of the centre, I, is determined by the position of L when the crank is on the centre, m falling at M_0 and at M_1, alternately, and the perpendicular at the central intermediate point locating the vertical through I; then $M_0I = M_1I = ML$.

* In this, we follow methods probably first employed by M. Boulvin of Liege; Ann. de l'Assoc. des Ingenieurs; 1884.

The ellipse of which the minor axis is ka is the path of M, and affords the means of finding the path of L. Neglecting obliquity, $Ml = M_0 l_0$ in Fig. 145, and lL, extended to l', locates the desired point. Or: measure $mM = l_0 l$; draw dd parallel to DD, and draw a triangle on II similar to LIl'. Then a horizontal drawn, at any instant, through M to the vertical, YY, gives the same measure, $l'L$, as an intercept between II and YY.

To determine the motion of the point S from that of M and of L, we need only observe that the ratio of SL to MS is constant, and make the construction below, in which μ and λ give the displacements of M and L from their mean positions, on MM and LL; an arbitrary scale also gives the division of SS in the ratio $\frac{m}{n}$. The new scale σ may be used in laying down the distribution.

Adopting the Zeuner method:

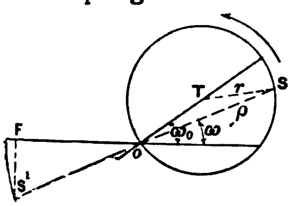

FIG. 146.—WALSCHAERT GEAR.

$$\mu = om = a \cos \alpha;$$

$$\lambda = ka \sin \omega \tan \alpha;$$

$$\sigma = \frac{m}{m+n}\lambda + \frac{n}{m+n} a \cos \alpha. \quad (1)$$

In Fig. 146 we have

$$\rho = 2r \cos (\omega - \omega)_0$$
$$= 2r \cos \omega \cos \omega_0 + 2r \sin \omega \sin \omega_0. \quad . \quad . \quad . \quad (2)$$

Since

$$r \cos \omega_0 = \frac{1}{2} \frac{na}{m+n},$$

$$= \frac{1}{2} \frac{m}{m+n} ka \tan \alpha,$$

we find

$$\tan \omega_0 = \frac{m}{n} k \tan \alpha; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$r = \frac{1}{2} \frac{a}{\frac{m}{n}+1} \sqrt{\left(\frac{m}{n}\right)^2 k^2 \tan^2 \alpha + 1}; \quad . \quad . \quad (4)$$

and it follows, as stated by Boulvin, that the motion of the valve is identical with that of another driven by an eccentric having the angular advance $180^\circ - \omega_0$.

In the usual form of gear the motion is determined by the angular advance and the eccentricity of the driving eccentric.

In this gear it is due to the magnitudes of $\frac{m}{n}$, k, and α. In Fig. 146, a movement, OF, of the piston, to the left, corresponds with that, of the valve, toward the right, which is measured on $S'O$. The coördinates of T are

$$x = r \cos \omega_0 = \frac{1}{2} \frac{na}{m + n}; \quad . \quad . \quad . \quad . \quad (5)$$

$$y = r \sin \omega_0 = \frac{1}{2} \frac{m}{m + n} ka \tan \alpha; \quad . \quad . \quad (6)$$

whence

$$y = \left(\frac{a}{2} - x\right) k \tan \alpha, \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

which equation holds good for the location of T, in Fig. 147, whatever the value of $\frac{m}{n}$; that point always being found in a straight line of which that is the equation, and which line passes through a pole, Q, so situated that $OQ = \frac{1}{2}a$; and which the angle with OQ is such as to have the tangent $k \tan \alpha$.

The point R is found by taking $\overline{OQ'} = k\,\overline{OQ}$, and $QQ'R = \alpha$; since, making, in (7), $x = 0$,

$$OR = OQ' \tan \alpha = k \tan \alpha \overline{OQ} = \tfrac{1}{2} ka \tan \alpha$$

From (5),

$$x = ot = \tfrac{1}{2}a \frac{n}{m + n},$$

and

$$OQ = \tfrac{1}{2}a; \ \frac{ot}{tQ} = \frac{n}{m}.$$

Thus the position of S on ML fixes that of the vertical through T; and if constant, variations of $k \tan \alpha$ only displace

T in this vertical line. The value of k must be so chosen that the angle of α shall not be too great.

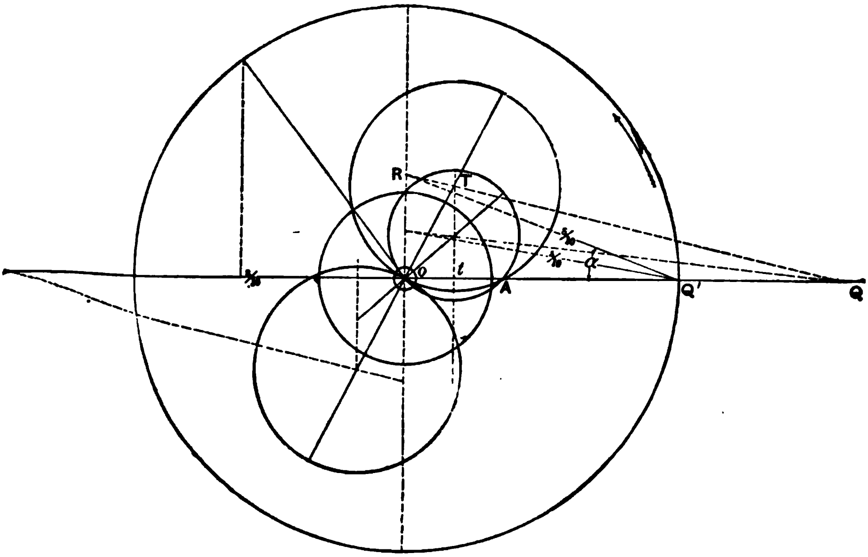

FIG. 147.—WALSCHAERT'S GEAR.

Thus to determine the motion of the valve, in any case, take $OQ = \frac{1}{2}a$: $OQ' = k\overline{OQ}$; let $Q'R$ make an angle α with the horizontal; join QR, and the centre of the valve-circle is at the intersection of QR with the vertical through t, and t divides QO into two parts having the prescribed ratio, $\frac{m}{n}$.

Swinging the guide DD past the vertical, into the corresponding opposite inclination, places the gear in position for backward movement. All constructions are then symmetrical with those which have just been given.

To operate the Meyer expansion-valve, Fig. 148, take a point nearer M than is S, and divide the link MN in the ratio $\frac{m'}{n'}$. Two points T_1 and T_1' being taken, take $o\theta = t't$ and $o\Theta'$ parallel to $T_1'T_1$. The closure of the port corresponds to the position, ON, of the crank, for a negative lap η. A fortunate

incidental advantage of this arrangement is the production of free admission at short cut-offs.

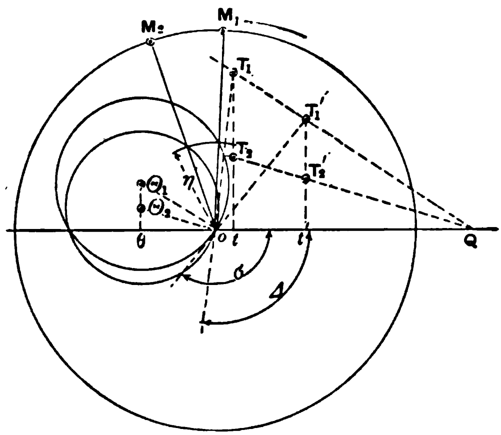

FIG. 148.—MEYER GEAR.

The Hackworth and Marshall gears may be similarly treated. In Fig. 149, the link ML receives its motion from the eccentric OM, of which the radius r is placed opposite the crank. In this case $a = e$, and k is given any appropriate value. $OQ = \frac{1}{2}e$; $OQT = \alpha$ (Fig. 151). To find the point T, as in Fig. 151, and similarly proportioned, the guide or directrix must be given considerable inclination; and the Marshall system gives an improved action when its method of suspension is used in place of the link.

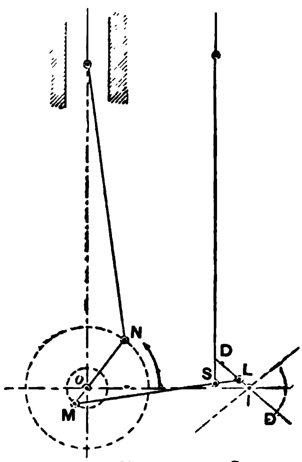

FIG. 149.—HACKWORTH GEAR.

In Fig. 150 the eccentric is set with the crank, and S' is found on $M'L'$. To obtain this construction, take in Fig. 151 at the left $OQ' = \frac{1}{2}e'$; e' measuring the eccentricity. Take

$$\frac{to}{tQ'} = \frac{S'L'}{S'M'}.$$

and $Q'T$ must make the angle α with the horizontal. With similar angle of direction and

$$\frac{S'L'}{S'M} = \frac{SL}{SM'}$$

the eccentricity may be made less; and we may obtain

$$\frac{OQ'}{OQ} = \frac{m-n}{m+n}.$$

Valve-motions of these kinds often illustrate great lightness and compactness, and usually with few parts. They may all be made to give uniform lead. The disuse of the eccentric is an important point, especially at high speeds of rotation.

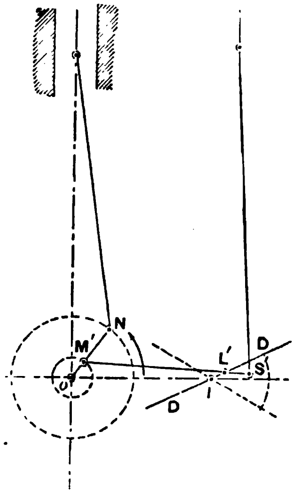

FIG. 150.—MARSHALL GEAR.

75. Steam Reversing-gears are required with all very large engines; as it is quite impossible to reverse the valve-

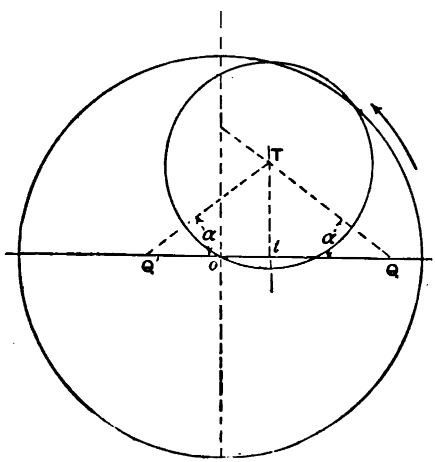

FIG. 151.—MARSHALL GEAR.

gear, to stop, to start, or to reverse the motion of the engines by hand conveniently or quickly. There are many forms of

this apparatus, some of which are designed to actuate precisely such arrangements of reversing-levers or other devices as are customarily moved by hand; while others are independent and special contrivances to effect the same purpose. In one class of proved value the reversing-engines are a pair of steam-cylinders driving a crank-shaft on which is a worm which engages a gear on the "tumbling-shaft," actuating the link and throwing it one way or the other as the reversing-engines themselves are worked in one or the other direction. In another form, adopted by Messrs. Randolph & Elder about 1855, the reversing-engines are so attached that the main engines may be stopped or reversed by the movement of the former, driving the main working eccentric ahead on the shaft until the engines are stopped or reversed as may be desired. In still other cases, as on locomotives, for example, the reversing-engine consists simply of a steam-cylinder with its piston-rod directly attached to the reversing-lever or its connections. In such cases some system of regulation is needed to prevent its jumping the full length of its stroke or working irregularly; this is sometimes prevented by a controlling screw on the piston-rod or some convenient connection. Steam is turned into the cylinder and the screw then turned by hand, as if the latter were supplying the working power, although it simply regulates the motion produced by the force of the steam acting on the piston of the reversing-engine. The most satisfactory direct-acting system is one in which the valve of the reversing-engine is operated by a hand-lever, the fulcrum of which is secured upon its cross-head or piston-rod, and in such manner that the motion of the latter, the hand not moving, will close the valve. The effect in actual operation is then to permit the motion of the engine to follow that of the hand, and never to allow the engine to jump or to suddenly move the main valve. With this arrangement the largest engines are as easily moved and with as great precision as the smallest.

CHAPTER III.

THE REGULATION OF THE ENGINE.

GOVERNORS; FLYWHEELS; INERTIA AND ROTATIVE EFFECTS.

76. **The Disturbing Forces** in the steam-engine are either external, internal, or, as is generally the case, both within and without. The variations in the speed may be due to changes in the impelling forces, the load remaining unaltered; as when the steam-pressure varies, or when the distribution of the steam is changed; or those variations may be a consequence of changing load without corresponding alteration of the energy expended by the steam. These variations may also be either confined to the action of the engine, revolution by revolution; or they may occupy less time than a single revolution, and be observed as a fluctuation of the speed of the fly-wheel as it turns; the number of revolutions, and its mean velocity, minute by minute and hour by hour, remaining absolutely constant.

Of these several kinds of disturbance, the first is seen most usually where an engine without a governor, as a marine engine, works against a steady resistance and the boiler-pressure, or the position of the throttle-valve, is changed; the second is seen best in a rolling-mill; the third is best illustrated, perhaps, in the action of a locomotive engine; the fourth in any case of moderate speed of revolution when the engine has a light fly-wheel, or, as sometimes is the case with coupled engines, none at all, or when the ratio of expansion adopted, regularly or momentarily, is too high. In the latter case the effort on the piston greatly exceeds the normal resistance at one part of the stroke and is correspondingly less at another. Steam pumping-engines and marine paddle-engines also often exemplify this case.

The Principle of Regulation may be stated simply as a paraphrase of Newton's Laws. These laws lead to the deduction that, to secure absolutely uniform motion, perfect equality must be secured between the opposing driving forces and load at every instant; that every minutest change in the resistance must be accompanied by a precisely equal and simultaneous change in the steam-pressure or the power of the engine.

Absolutely exact apportionment of driving power to driven load is impracticable in the steam-engine; but it may be possible, in any given case, to reduce variation of force and of speed to within any required limit; and such variations of speed are actually restricted to exceedingly narrow bounds. This is done by varying the quantity of steam used, at each stroke, to give the desired power, and by applying a fly-wheel to store and restore variations of energy occurring during each stroke or revolution.

77. **External Variations of Energy** or of load are consequent upon changes in resistance due to the throwing on and off of machinery, as in mills and factories, the changes of load in machinery in continuous motion, as in rolling-mills, or, in locomotives, to the continual variation of power demanded to haul a certain weight of train over a track alternately rising and falling, or of changing curvature.

It is ordinarily impossible to render these kinds of load steady, or even approximately so; and the three expedients which may be resorted to are:

(1) The addition to the regular load of another load is effected; which latter may be varied at pleasure in such manner as to make the total load constant. This is the process of regulating an engine or other motor by the use of a brake. It is a wasteful method, and is resorted to only in special cases and for special reasons.

(2) The usual expedient is the regulation of speed by adjusting the power to as nearly as practicable exact equilibrium with the load. This is done by the governor, and is effective and economical.

(3) The fluctuating variations of equilibrium between power

and load are taken up and given out by a reservoir of energy. This reservoir is the fly-wheel, and is especially efficient in controlling those violent and sudden variations which the governor cannot be relied upon to meet satisfactorily. The fly-wheel evidently can act efficiently only where the mean power and mean load are balanced, and its storage and restoration of energy thus rendered, on the whole, equal.

78. **General Internal Variations of Energy** and power in the steam-engine, producing changes of speed with a constant load, are due to such variations of effort as follow the gradual changes of steam-pressure at the boiler and in the steam-chest. Such changes of speed as follow are only controllable by a governor so constructed that it shall produce its effects through variations of its own speed. The greater its sensitiveness to such variations, other things equal, the more accurately will the speed of the engine be preserved. These general variations of power being reduced to zero, the mean speed of the engine, minute by minute, will, with constant load, be uniform.

79. **Momentary Internal Variations of Energy** are due to two causes: (1) the variations in the effort of the steam acting on the piston, consequent upon admission, expansion, exhaust, and compression; (2) the irregular application of pressure on the crank-pin coming of the harmonic kinematic relations of the piston and shaft motions, and of the disturbances due to the vibration of the connecting-rod, and the still other and superadded fluctuations of effect produced by the unsymmetrical motions of the rod on either side the position of maximum velocity of the piston. Such changes of effort cannot be controlled by the governor, and must always be the subject of regulation by the fly-wheel; or, in special cases, as in the modern Worthington engine, by special equilibrating devices.

80. **The Fly-wheel and its Work** must evidently be studied from two distinct standpoints: (1) the external variations of load, so sudden and violent as to be beyond control of the governor, and affecting the speed of the engine in less periods of time than a single stroke; (2) those internal changes

of pressure on the piston and crank-pin occurring within the same restricted period of the engine-cycle. No wheel is adequate to its purpose which has not energy-storing power sufficient to meet either of these independent variations, and even both together, under the most unfavorable contingencies anticipated by the designer or the user of the machine.

The wheel may be called upon to act as a regulator to either the driven machinery, or to the driving engine, or to both at once; and the designer cannot be certain of successful regulation until he has ascertained the probable magnitude of both these fluctuations, and provided for their control when acting separately or when their effects are superposed. In all cases the fly-wheel is seen to be that element of the engine which is relied upon to reduce momentary fluctuations of speed to within an allowable range.

81. The Governor and its Action are obviously not to be taken as substitutes for the fly-wheel and its effects. The one may, and usually does, act in conjunction with the other to control fluctuations of engine-speed; but each has its special place and purpose. As the fly-wheel is expected to keep all variations of velocity of rotation, within the single revolution, closely approximated to a certain required mean, the governor, on the other hand, must keep that mean speed constant, revolution after revolution, throughout the period of operation of the engine, by keeping the steam-supply constantly and precisely adjusted to the requirement of the load. The governor fixes the mean number of revolutions of the engine, under all circumstances, and holds the engine to it more or less closely; the fly-wheel keeps the speed closely to the mean of each revolution, instant by instant, throughout such revolution.

82. Inertia-forces, and their Effects, assume importance where the speed of piston and of rotation become great. In the days of Watt, with these speeds rarely exceeding 200 feet, and, in large engines, ten or fifteen revolutions per minute, such forces required no consideration in theory or in practice. In recent times, with speeds rising above one thousand feet, and with even the largest marine engines approaching a hun-

dred revolutions per minute, their magnitude is a matter of real importance. Such engines, when their speeds are increased, by any accident, to three or four times their usual magnitudes, may go to pieces through the action of inertia of their heavy parts. In all "high-speed" engines, in which, often, the speed of rotation exceeds two hundred revolutions per minute in the larger, and is above 300 in the smaller, sizes, the inertia of their reciprocating parts may give rise to forces of great intensity, forces which may considerably modify the distribution of stresses throughout the engine.

On studying these forces, their action in changing the normal pressures on the crank-pin will be found to be the most interesting, and perhaps most important, of their effects. This action was considered by Mr. Porter so advantageous in engines having high speed of rotation that he added weight to the piston and cross-head to secure that comparatively uniform pressure which he discovered to be thus obtainable.

83. **The Brake, as a Regulator** simply, has comparatively little importance to the engineer. It is always an apparatus which wastes energy by its conversion from the dynamic into the thermal form, and by its dispersion in such a manner that it can never be regained or utilized. The brake is, for this reason, never employed for the regulation of continuously operating engines, and is reserved for the purposes of the engineer testing engines and needing, for this purpose, a power-absorbing dynamometer; or where, as in hoists and for similar applications, energy is not utilizable by storage and must be wasted.

84. **Special Regulating and Equilibrating Devices** have been often employed as regulators and oftener proposed or only experimentally and unsuccessfully used. Of the latter, and perhaps the earliest of the class, was the invention of Watt, which equilibrated the changing pressures due the expansion of steam by a corresponding displacement of the centre of suspension of the beam and consequent alteration of lever-arm in such manner as to give an approximately constant effort at the opposite end of the beam and at the crank. In this category,

also, falls the attachment to the "high-duty" engine of Mr. Worthington; where, no fly-wheel being used to adjust pressures and resistances, no high ratio of expansion could be adopted, and no great efficiency attained, except by the use of some such plan. They are of too exceptional use to necessitate detailed study here. (See Part I, § 37, p, 164.)

85. **The Office of the Governor** is to adjust the steam-supply exactly to the requirements of the engine, making the mean effort during each revolution precisely equal to the mean resistance, the essential condition of uniform speed. This result may be attained in either of two ways, in ordinary cases: the influx of steam into the steam-chest may be regulated by the operation of "throttle-valve;" which is substantially equivalent to variation of the boiler-pressure; or the governor may determine the point of cut-off, and ratio of expansion; in either case determining the amount of work performed at the piston of the engine, stroke by stroke.

The governor is commonly an apparatus in which, at the desired and normal speed, an equilibrium exists between two opposing forces; which equilibrium is destroyed by alteration of speed of engine; the force then in excess acting to change the position of the valve or other apparatus by which speed and power are adjusted to the momentary demand. One of these forces is most frequently that due to the action of inertia in some revolving mass, to centrifugal force; and the opposing force is that of gravity. This is the principle of the common "fly-ball governor" of Watt, a revolving pendulum. In some cases, the counterbalancing force is exerted by a spring; in a few forms of governor, the resistances of a vane, moving in a fluid and a weight or spring balancing it, constitute the applied forces.

Variation of speed takes place, as has been seen,

(1) Whenever the resistance varies;

(2) When the power varies;

(3) When both vary in different proportions;

(4) When the efficiency of the system varies.

In all cases, the office of the governor is to adjust the power to the demand and to thus preserve a perfect balance.

Governors are said to be *static*, or to possess stable equilibrium, when they maintain a definite position at any given speed, and resist external forces tending to alter their "conformation." They are in unstable equilibrium or *astatic* when they assume indifferently any position at a certain speed, and offer no resistance to alteration of their "conformation." In the latter case, there is but one speed at which the apparatus is in equilibrium, and any change of speed from this normal velocity will cause the instrument to exert an effort to restore the conditions producing regular and normal speed. This class of governor is that more generally denominated *isochronous* (Gr. *isos*, equal; *chronos*, time). Pseudo-astatic, or pseudo-isochronous, governors are those oftener called "approximately isochronous."

86. The Essentials of Good Regulation are:

(1) Promptness of action;
(2) Accuracy of operation;
(3) Delicacy of adjustment.

These qualities are secured by obtaining maximum power in working, minimum friction, in the governor and its connections, light load and ease of movement at the point where its work is performed, and freedom from irregularity in resistances. Increasing the speed of rotation and weight of balls, for example, in the common fly-ball governor, and reduction of resistance, such as is observed when comparing the Corliss or the Greene with the single-valve, "automatic," forms of valve-gear, or with throttling systems, are common illustrations of this case. The heavier the governor and the less its load the more prompt its action. The method of attachment usually determines the accuracy of operation and the delicacy of adjustment. In many cases, irregularities due to sudden fluctuations and oversensitiveness of the governor become serious, and special devices must be introduced to secure smooth working.

A good governor must thus be:

(1) Isochronous, or nearly so;

(2) Sensitive;

(3) Powerful, and, hence,

(4) Prompt and exact in its action;

(5) Compact;

(6) Light as possible;

(7) Convenient of attachment and operation;

(8) Safe against accident, and against liability to cause accident;

(9) Simple in construction;

(10) Inexpensive.

87. **The Classification and Definition** of governors may be made on either of several principles:

According to purpose, as:

Engine-governors;

Stationary: $\begin{cases} \text{throttling;} \\ \text{adjusting cut-off;} \end{cases}$

Marine;

Water-wheel governors;

Clock-work governors.

The last named are called escapements.

The marine governor has usually quite a distinct form and purpose, in some sense, from the others, and has been called the "fly-governor."

The familiar classes of governor may also be classed *according to method of action*, as:

Position-governors;

Disengagement-governors;

Differential governors.

Governors are also classed, *according to form*, as:

"Fly-ball," or pendulum, governors;

Spring governors;

Fluid governors ·

Pump;

Bellows;

Fan;

Chronometric governors;
And the first are either:
Simple pendulum governors; or
Loaded pendulum governors.

Governors are defined, as above classed, as follows:

Engine-governors are those, of whatever construction, which are employed to regulate and control the speed of steam or other engines. The stationary engine usually requires a governor having all the qualities ascribed to the best types. Cotton-mills making fine goods, and electric-lighting apparatus, demand extremely regular speed; but saw-mills, rolling-mills, machine-shops, and many kinds of manufacturing establishments require less nicety of regulation.

Marine engines are ordinarily regulated by hand; but, in a heavy sea, the pitching of the ship is often so great as to throw the screw entirely out of water at one instant, and to bury it deeply in the sea at another; thus causing the most sudden, violent, and sometimes dangerous, fluctuations of speed. It is to prevent these peculiar variations that the marine engine, or fly, governor is applied. The locomotive engine is regulated by hand, and no governor is attached.

With water-wheel governors the power required to move the gate promptly and satisfactorily is too great to permit the direct application of the governor to regulation, as is usual in steam-engines; and it is therefore here used to throw in and out of action trains of gearing or belted pulleys driven by the wheel itself, and so connected as to open or to close the gate. Such types of governor are, occasionally, also applied to the steam-engine.

Clock-work governors, or escapements, are employed, not to secure a steady movement at a stated rate, but to insure perfect regularity of intermission. They hold the clock movement at rest for a definite period, then allow the controlling piece to escape for an instant, and a definite movement to take place. They then again hold the whole at rest for the desired period of time. It is this intermittent action which gives them the horologist's name, escapement.

Position governors are those with which the piece, valve or other, which controls the movement of the motor is so connected with the governor, and so moved by the latter, that the position assumed by the whole train is determined by the governor. This is illustrated by the two common forms of steam-engine governor with which, in the one case the throttle-valve, in the other case the expansion-gear, or mechanism, is directly moved by the governor; the one determining the area of opening in the steam-pipe, the other fixing the distance to which the steam may follow the piston, or point of cut-off, and the ratio of expansion.

Disengagement-governors are those in which the connection with the regulating mechanism is not fixed; but, the machine moving at its intended speed, the governor revolves entirely free; the speed increasing beyond the fixed limit, connection is made with a train of mechanism which closes the regulating valve; or, speed falling too low, with a train opening that valve. Regulation is thus effected by this alternate engagement and disengagement, as in the water-wheel governors.

Differential governors are those with which a constant and standard speed is established, independent of the speed of the machine to be regulated; and, whenever the latter departs from its proper speed, the difference between its velocity and that of the governor is made to produce such an action on the regulating valve or mechanism as will tend to restore the correct speed.

The fly-ball, or common pendulum, governor consists of a pair of balls, or other heavy weights, suspended by rigid arms from a spindle, in such manner that, as this spindle revolves, the balls assume orbits more or less remote from that axis their arms being driven, usually, from the spindle through joints which permit their motion in the vertical plane. In such governors, centrifugal force tends to cause the balls to separate, their arms diverging from the spindle; while the action of gravity tends to restrain this movement and, if the instrument is properly proportioned and connected with the engine, an equilibrium can occur only when the speed is correct. This

balance being effected by the adjustment of the regulating mechanism in such manner only as will supply the working fluid in such quantity as to drive the engine at the intended speed, the constant velocity so obtained is that required to do the work of the machine at exactly, or very nearly, the desired rate.

Spring governors are those in which the action of gravity is replaced by that of a spring. Since gravity is constant, and the action of a spring varies with its movement, it is evident that the latter is controlled by a different law, and has its own theory.

Fluid governors are those in which some fluid supplies a resistance which equilibrates the action of an independent force, as that of gravity or of a spring, when the speed is constant. Each form has its own method of operation and its own theory.

Loaded governors are those in which a weight or an accessory spring is connected with the simple mechanism of the apparatus, so as to permit an increased speed to be attained at equilibrium by checking the effect of centrifugal force or its substitute, and the range of movement of the balls, at any given speed.

88. **Designing the Governor** is a process involving a knowledge of the nature of the motion to be controlled, the resistances acting upon the governor, and the general construction and action of the instrument itself. Generally, the simpler the construction, the higher the maximum allowable speed, and the more positive the system of attachment, the safer and better its working; the less the resistance of the mechanism operated by the governor, the more prompt and efficient its action.

In designing, therefore, the effort should always be made to secure simplicity, positive connections, freedom from friction, great power, and the least possible action on the governor by the regulating mechanism which it controls. In all cases, the governor, at normal speed, is in a position of equilibrium, and its effort becomes greater or less as it is forced away from, or approaches, this position, which is one of absolutely no power; and thus, the less the resistance to its action, the more exactly

will it hold the engine to speed. Governors are efficient as their energy is greater and as this resistance is less. The governor should be connected to the engine by a strong and positive train of mechanism, and in such manner that risk of breakage shall be slight, and that, in case of such accident, the engine shall stop. Broad bearings, so arranged that no binding or jamming can be caused by any sudden variations of speed, and free and certain lubrication, are important details to be studied in designing. A detaching device intended to act, should the belt break, is often introduced with the belted governors.

In some special cases, as in most marine governors, which are intended to prevent sudden or extreme fluctuations of speed, rather than to secure perfect uniformity of motion at a stated rate, the device consists of a revolving mass, driven by a flexible and elastic connection, in such manner that, when a quickly occurring change of speed of engine produces a difference of motion and relative position between it and the connection, in consequence of its inertia, that relative displacement shall so move the regulator as to check the fluctuation of speed.

In all governors depending upon the action of freely moving bodies, as in the common revolving pendulum, or fly-ball, governor, and in the marine governor just referred to, the power with which the governor acts is proportional to the energy of those parts; i.e., to the product of their weight and the difference of the squares of the velocities. The work to be done by the governor is also measured by the product of the resistance which it is to overcome into the space through which it is met during a change in the "configuration" of the governor corresponding to the allowable variation of speed. These two products are thus equal, in every governor, and if R is the resistance, s the space through which it is expected to act, and if W is the total weight of the acting parts of the apparatus, as the balls of the common governor, v_1, v_2, their velocities, and s' the space through which their effort acts while the permissible variation is taking place,

$$Rs = \frac{W(\Delta v^2)s'}{2g}.$$

We may therefore take this energy to represent the power of any governor; and the value of the apparatus is determined by this measure, by its sensitiveness, and by its isochronism.

Every governor must be connected to the regulating valve through a train of mechanism having the least possible friction and in perfect balance. As the governor, at normal speed, is in a state of perfect equilibrium, it can only exert power by the disturbance of that equilibrium and in a degree which is the nearer zero as that condition is the more exactly attained. That governor is the most efficient also which, having the needed power and sensitiveness, is of smallest dimensions and has least weight. That system of regulation is best, the governor being satisfactorily efficient, which imposes least work on the governor.

89. **In Construction of the Governor** it is essential that all its parts should be carefully fitted, all joints and all journals and bearings well made, and every piece exactly balanced with its opposite, in such manner that the governor may work with perfect smoothness and freedom from jar. When the governor is weighted—as the Porter governor, for example—the weight, especially if heavy, should be so set that it may turn freely, and without friction, upon the piece which carries it. A washer, or, better, two or three, each lubricated on top and bottom, forms a good support. If driven by a belt, the latter should be wide enough to drive easily and without possibility of slipping, however slack; if driven by gearing, it should be carefully cut and of ample size of tooth. No rubbing, or noise due to backlash, should be observable.

90. **The Attachment and Operation** of the governor, as has been seen, demand attention from the designing engineer. If the governor moves a throttle-valve, and thus determines the pressure of steam in the valve-chest and the steam-pressure reaching the engine, it must be proportioned and attached not only so as to give ample power, but also so that the valve may be moved promptly, strongly, and throughout the full range of action demanded. The more direct, simple, and frictionless the system adopted the better. The higher the speed at

which the governor is designed to be operated, the more effective will it be; the nearer it is placed to the source of its motion, and the shorter its connections with the regulating mechanism, other things equal, the better its working.

When the governor is connected to a throttle-valve, it is necessary to determine how great a fluctuation of speed shall completely close the valve, on the one hand, or open it wide on the other. Too wide a range causes trouble in consequence of the irregularity of speed wasting power, making machinery inefficient, and sometimes causing serious losses; while too narrow a range causes difficulty by destroying sensitiveness. Bringing in, as a new element, the vibrations of the governor itself, due the inertia of its parts, a similar effect is produced by that cause; which must be compensated by the use of a "dash-pot." There is usually a range, best for the particular combination used, and which can only be satisfactorily determined by experiment.

91. **The Governor and the Throttle-valve** form the system of regulation in most common use in the oldest types of engine. As a general rule the fly-ball governor of Watt is adopted; and it is driven by a belt from a pulley on the crank-shaft; the governor being placed at any convenient location between main shaft and throttle. Bell-cranks and "reach-rods" connect it with the arm of the valve; the whole arrangement being substantially that originally adopted by Watt. In such cases no very exact regulation is usually possible with even the best designing and construction. The wide range of action demanded of the governor, and the great friction of the mechanism, preclude efficient operation. Heavy balls, long arms, and low speed of rotation are the characteristics, usually, of this system, and if a variation of speed of engine as low as five per cent, under trying circumstances, can be attained, the result may be accepted as very satisfactory.

In the more recent and better class of throttling governors, the valve is set directly on the steam-chest of the engine, and a spring or loaded governor is driven at high speed, mounted on the valve-casing, in such manner that the spindle of the former is in line with the valve-stem, and the connection is

very short. A balanced valve and a very slender stem insure minimum resistance to motion. Such a system is often constructed by independent makers and sold to the builders of the engine. They generally include a safety detaching mechanism. With this type of governor, if the space between governor-valve and engine-valve can be sufficiently reduced in volume, fairly good regulation may be secured.

92. **The Governor and the Link-motion** constitute the regulating mechanism of several types of engine. In this case, the work of the governor is the raising and depressing of the link, to vary the lead and travel of the valve in the manner described in the chapter on valve-gearing. The link, in such cases, should always be so balanced as to prevent any excessive or irregular resistance affecting the governor.' The friction in the link itself, when moving, need not be great; but the work of the governor is, nevertheless, considerable, since it must usually act through a train of intermediate mechanism of considerable weight and inertia, moving a heavy mass of metal through a somewhat wide range. Here, therefore, the regulation is often less satisfactory than in the preceding case, and a spring or weighted governor is here a still more essential element of successful regulation. With a loaded governor, high speed, a balanced system of valves, and carefully counterbalanced link, it is practicable to secure good results.

93. **Governors actuating Automatic Gear** are illustrated in the design of many of the best engines in use. In such cases, their purpose is to determine that point of cut-off, and that ratio of expansion, which will precisely adjust the power exerted at the piston, at each stroke, to the amount demanded by the work, by giving the valve the needed variation of lead or of travel, or of both. Where, as is common, this is effected by shifting the eccentric around the shaft or across it, the governor is usually itself mounted on the main shaft, and directly connected with the eccentric.

In these forms of gear, the work of the governor is necessarily somewhat heavy, even when handling only the eccentric; and the valve must be well designed, and always so balanced as to make the total resistance a minimum. The whole

system should be in both static and running balance; otherwise it will be impossible to avoid a tendency to oscillation, and consequent fluctuation of engine-speed. As this class of governor cannot work well at low speeds of rotation, this arrangement can only be adapted satisfactorily to high-speed engines. Speeds of 150 to 200 revolutions per minute may be taken as minima. For the governor itself, considerable range of movement is essential, as well as strength, sensitiveness, and stability sufficient to prevent oscillation. This governor is usually placed in the driving-pulley, or fly-wheel, which then serves as its frame and carrier.

94. **Governors applied to the "Trip Cut-off" Gear** are commonly of the gravity type, with vertical spindle, and are often loaded. The best varieties of drop cut-off gear are so designed and constructed that they demand no other work of the governor than to detach a part, to disengage a catch or a latch, or to otherwise effect the tripping of the steam-valve or cut-off; and this is so little as to affect the action of the governor quite imperceptibly. The designer has here, therefore, greater liberty, in choice of type, and in design, than in other cases. These governors are very usually belted, and must be provided with safety devices, to operate if the belt should slip, break, or be thrown off.

95. **The Theory of the Revolving Pendulum,** used as a governor is, in its simplest form, as follows:

Centrifugal force is the name applied to that effort which, in a body revolving about a centre, tends to break the cord or arm holding it at a definite distance from the centre of its orbit. This effort is due to inertia and the tendency of every moving body endowed therewith to continue its motion, according to Newton's law, in a straight line and with unvarying velocity. Every attempt to change its motion is met by a resistance on the part of the mass which is precisely proportional in amount to the rapidity of that

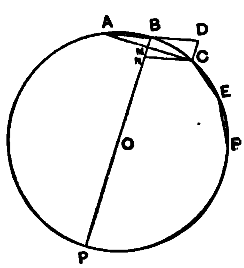

FIG. 152.—CENTRIFUGAL FORCE.

change and which is felt in a direction precisely opposite to that of the change of motion. Thus, in Fig. 152, let ABC be the path of the mass and let it describe AB with the velocity v. If, when at B, no other forces act, it will describe $BD = AB$ in the same time. Join DC, and draw BN parallel to DC. If, when at B, the point receive an impulse in the direction BN, such as to cause it to describe BN in the same time as BD, the mass will describe BC. But since $BC = BD$, the velocity in BC is the same as that in AB. Since the triangle BCD is isosceles, $BDC = BCD = CBN$. But $ABN = BDC$; $\therefore ABN = CBN$, and BN bisects the angle ABC. Hence the direction of the impulse will be toward the centre of the circle.

Then

$$BM = \frac{\overline{BC}^2}{2r}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and

$$f = BN = \frac{\overline{BC}^2}{r} = \frac{v^2}{r} = a^2 r; \quad . \quad . \quad . \quad (2)$$

when a is the angular velocity; and the measure of centripetal, as of centrifugal force is the square of the velocity divided by the radius of the circle; or the angular velocity squared multiplied by radius; for when the number of sides becomes infinite, or the polygon in the figure becomes a circle, the succession of impulses will become an incessant action of the same intensity.

The force is a constant accelerating force.

Then if v is constant, $f \propto \frac{1}{r}$; if r be constant, $f \propto v^2$ and $f \propto a^2$; and if the angular velocity, or the number of revolutions in the unit of time, be constant, the value of f varies as the radius of the orbit of the weight.

Since we always have

$$\frac{F}{W} = \frac{f}{g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

when F is the statical, or weight, measure of the accelerating effort,

$$F = f\frac{W}{g} = \frac{Wv^2}{gr} = \frac{W}{g}a^2r \quad . \quad . \quad . \quad . \quad (4)$$

The space described in one revolution, made in the time $T = \frac{1}{N}$, is

$$vT = 2\pi r; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$v = \frac{2\pi r}{T} = 2\pi rN = ar; \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$f = \frac{v^2}{r} = \frac{4\pi^2 r}{T^2}; \quad F = \frac{4\pi^2 rW}{gT^2}; \quad . \quad . \quad . \quad . \quad (7)$$

and the value of f or of F varies as the radius multiplied by the square of the number of revolutions per second.*

It is evident that this accelerating force is to the force of gravity as twice the height due the velocity in the orbit is to its radius.

There exists at every position, if the governor revolve with uniform motion, an equilibrium between the vertical component of the force acting along the suspending arm, AB, the force of gravity, and centrifugal force. The height of the point at which the line of the arm crosses the vertical spindle,—which is the virtual point of suspension,—above the plane of revolution of the balls, bears a ratio to the radius of the orbit in which the balls move, precisely equal to the ratio of the weight of ball to the effort of centrifugal force, *i.e.*, Fig. 153.

$$t \sin\theta - \frac{mv^2}{r} = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$t \cos\theta - mg = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$v^2 = \frac{gr \sin\theta}{\cos\theta};$$

$$= gAB\frac{\sin^2\theta}{\cos\theta}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

* It is obvious that, in the above expressions, all measures are taken in the same terms as g, *i.e.*, feet if in British measure, meters if in metric.

$$\text{and } T = \frac{2\pi r}{v};$$

$$= 2\pi\sqrt{\frac{AB \cos \theta}{g}};$$

$$= 2\pi\sqrt{\frac{AC}{g}} = 2\pi\sqrt{\frac{h}{g}}; \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and the speed and altitude are independent of the weight of the balls, friction resistance being 0. The number of revolutions per second $N = \sqrt{\frac{0.815}{h}}$; and the height of the point of vertical suspension above the plane of revolution of the balls, $AC = h = \frac{0.815}{N^2}$ feet $= \frac{9.748}{N^2}$ inches, $= \frac{0.248}{N^2}$ metres.

Also $h = \frac{35208}{R^2}$, where h = height in inches, the number of revolutions per minute, $R = \frac{187.5}{\sqrt{H}}$.

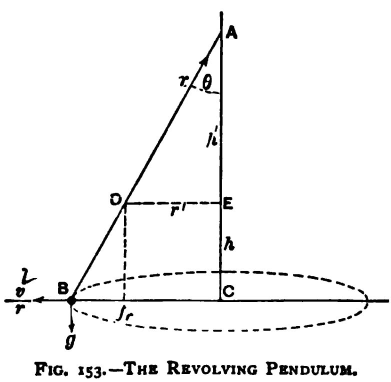

FIG. 153.—THE REVOLVING PENDULUM.

The weight of the balls usually so greatly exceeds that of the suspending arms in this class of governor, and the approximation desired is so coarse, that the theory generally adopted,

as above, takes no account of the action of the arms. They may, nevertheless, exert, in some cases, an important influence, and, if required, their effect may be readily taken into account, thus:

In Fig. 153, let D be the middle of the arm, AB, measured from the point of suspension, A, to the exterior of the ball, B. The sum of the moments of the weights will be, if B = weight of ball and R that of rod,

$$\Sigma M = B \cdot BC + R \cdot DE,$$

and must be equal to the sum of moments of centrifugal force acting on ball and arm; or to

$$\Sigma M = B \cdot \overline{AC} \cdot \frac{4\pi^2 \overline{BC}^2 N^2}{g \cdot \overline{BC}} + R \cdot \overline{AE} \cdot \frac{4\pi^2 \overline{DE}^2 N^2}{g \cdot \overline{DE}}$$

$$= B \cdot h \cdot \frac{4\pi^2 r^2 N^2}{gr} + R \cdot h' \cdot \frac{4\pi^2 r'^2 N^2}{gr'} \quad . \quad . \quad . \quad (12)$$

But $dR = R\frac{dx}{a-b}$, where a is the length of ball-rod to the centre of the ball, and b is the radius of the ball.

Then,

$$\Sigma M = B \cdot h \cdot \frac{4\pi^2 r N^2}{g} + R \cdot \frac{4\pi^2 N^2}{g} \int_0^{a-b} \frac{h' r' dx}{a-b}$$

$$= B \cdot h \cdot \frac{4\pi^2 a \sin\theta \, N^2 +}{g} + \frac{4R\pi^2 N^2 \cos\theta \sin\theta}{g} \cdot \frac{1}{3} \frac{x^3}{a-b} \quad . \quad (13)$$

and

$$\Sigma M = Ba \sin\theta + R\frac{a-b}{2} \sin\theta$$

$$= \frac{4hB\pi^2 a N^2}{g} + \frac{4\pi^2 R N^2 \cos a \sin\theta \, (a-b)^2}{3g} \quad . \quad . \quad (14)$$

But $\cos\theta = \frac{h}{a}$, and

$$\Sigma M = 4\frac{h}{g}\pi^2 BaN^2 + \frac{4R}{3g}\pi^2 N^2(a-b)^2\frac{h}{a} \quad . \quad . \quad . \quad (15)$$

$$\therefore h = \frac{B\cdot a + \frac{1}{2}R(a-b)}{\frac{4\pi^2 N^2}{g}\left(B\cdot a + \frac{R(a-b)^2}{3a}\right)}. \quad . \quad . \quad . \quad (16)$$

The value of the height h, when the weight of rod is neglected, is

$$h_1 = \frac{g}{4\pi^2 N^2},$$

$$\therefore \qquad \frac{h}{h_1} = \frac{B\cdot a + \frac{1}{2}R(a-b)}{B\cdot a + \frac{1}{3}R\frac{(a-b)^2}{a}} = \frac{1 + \frac{R(a-b)}{2B\cdot a}}{1 + \frac{R(a-b)^2}{3Ba}} \quad . \quad . \quad (17)$$

and the true height is obtained by multiplying the height computed for the simple pendulum by this last quantity. The corrected height is obviously always greater than the uncorrected altitude.

Were the rods represented, in Fig. 153, by BD, and pivoted at D to a bracket DE, carried on the side of the spindle, the height for computations should still be taken as AC; for the equation of equilibrium must be

$$\frac{4\pi^2 B\cdot\overline{BC}N^2}{g\overline{BC}} = B\cdot\overline{Df}; \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$\overline{Df} = \frac{\overline{Bf}\cdot g}{4\pi^2 N^2\overline{BC}};$$

$$= \frac{g}{4\pi^2 N^2}\cdot\frac{\overline{Df}}{\overline{Ac}} = h\cdot\frac{\overline{Df}}{\overline{AC}} \quad . \quad . \quad . \quad . \quad (19)$$

$$h = \overline{Df} \times \frac{\overline{AC}}{\overline{Df}} = \overline{AC}, \quad . \quad . \quad . \quad . \quad . \quad (20)$$

as with the first described form.

In either case the value of h is independent of the weights, the latter being reduced to a common centre of gravity from which the altitude is reckoned.

The height of the governor, when of the simplest type, the revolving pendulum, has been seen to be

$$h = \frac{g}{4\pi^2 N^2};$$

where h and g are in the same measure and N is the number of revolutions per second. The following are the heights corresponding to speeds ranging from 60 to 600 per minute, or from one revolution to ten per second.

HEIGHTS AND SPEEDS OF GOVERNOR.

Rev. per Minute.	Altitudes.			Rev. per Minute.	Altitudes.		
	Feet.	Inches.	Meters.		Feet.	Inches.	Meters.
60	0.815	9.78	0.248	200	0.07	0.88	0.022
80	0.46	5.50	0.140	300	0.03	0.39	0.010
100	0.29	3.52	0.089	400	0.018	0.22	0.0056
125	0.19	2.25	0.057	500	0.012	0.14	0.0055
150	0.13	1.56	0.040	600	0.0075	0.09	0.002

It is here seen that the heights for high speeds, for speeds much exceeding 100 revolutions per minute, are too small to allow the instrument to be practically applicable to ordinary purposes, and that for such speeds the centrifugal force must be equilibrated by a greater force than the simple weight of the balls. It is thus that the loaded governors come into use; the altitude being forcibly increased by loading down the balls, usually by connecting them with a suspended weight, or by introducing a spring having the same purpose.

96. The Pendulum Governor, the conical pendulum, the centrifugal, or fly-ball governor, as it is variously termed, is the oldest of the engine-governors. It was invented by Huyghens about the middle of the seventeenth century, and used in the regulation of clock-work. Watt in 1784, and Hooper, in 1789, applied it to prime-movers, the one regulating his "newly-invented" steam-engine, and the other his windmills, by its action. This apparatus is shown in its simplest and oldest form in the accompanying sketch, as attached to the regulating or "throttle" valve of Watt's engine.

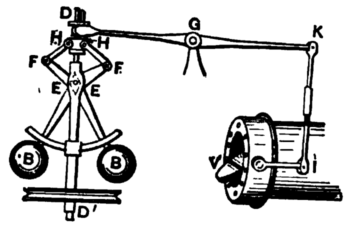

FIG. 154.—WATT'S GOVERNOR.

In this governor, the two heavy balls, BB, Fig. 154, are carried at the ends of the rods BF, which are suspended at E from the central spindle, DD, and through the latter driven by means of a belt or cord, on the pulley W, leading from the main-shaft or other convenient revolving part of the engine. The arms, BE, carry at their upper extremities shorter levers, EF, which, through the link-work parallelogram EFH, of which they form a part, move a collar, HH, and a lever actuated by it, HGK, connected with the throttle-valve in the steam-pipe at V. This line of link-work is so arranged that the separation of the balls with increasing speed closes the valve, and their fall toward the spindle, as the speed is reduced, opens it; thus increasing or decreasing the steam-supply, and always in such manner as to cause a return of the engine toward its proper speed, whichever the direction of variation. Nevertheless, as equilibrium can never be attained at the same speed for any two steam-pressures, or for any two loads, this governor is not isochronous, and it is impossible to obtain perfect constancy of speed with variation of either of these conditions. As the balls must rise as they swing outward, it always requires increased speed to hold them in the higher plane of revolution, and the engine must therefore work at higher speed when lightly than when heavily loaded.

97. **The Loaded Governor** is illustrated in Fig. 155, which represents the Porter Governor, a form which is superior to the preceding, the Watt Governor, in several details of construction. The addition of the weight, loading down the arms, as shown, permits driving the governor up to a higher speed, and thus securing greater quickness, sensibility, and power of action. The connection of the arms with the spindle by means of forked ends gives safety against pinching at the pins and resulting friction, and thus secures greater freedom of movement and increased delicacy. The weight is loose on the supporting collar, and its inertia does not interfere with the action of the governor by resisting sudden changes of its speed.

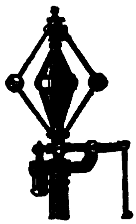

FIG. 155. LOADED GOVERNOR.

This governor is usually driven by a belt; but sometimes by gearing on the crankshaft. Whenever possible, a safety device should be introduced to stop the engine, should the belt or gearing break.

The Theory of Loaded Governors is as simple as that of the common pendulum governor, the weight of arms included. In this case, the moment of the weight of the balls, resisting the moment of centrifugal force, is reinforced by other weights so placed as not to be affected by the latter. These added weights are usually hung on the central spindle, but occasionally are carried on levers. The centrifugal action is thus due to that of the balls alone; while the effect of gravity is measured by the sum of the weights of the balls, and of the efforts of the added masses as felt at the balls. If the total weight of balls is B, and that of the weights, either as attached to and sliding on the spindle, as in the Porter governor, or as reduced to the equivalent weight so acting, if otherwise arranged, be W, the action of gravity is increased in the proportion of 1 to $1 + \frac{2W}{B}$; since the added load, thus connected, rises or falls through twice the altitude moved through by the same weight in the balls.

In Fig. 156, let $q = \dfrac{\overline{AD}}{\overline{AB}}$; then

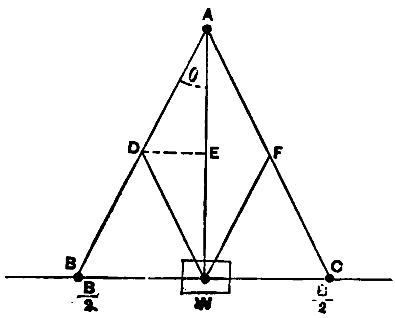

FIG. 156.—LOADED GOVERNORS.

$$B \cdot \overline{BW} + \overline{W} \sec\theta \cos\theta\, \overline{EW} + W \sec\theta \sin\theta\, \overline{AE}$$
$$= 4\frac{B}{g}\pi^2 N^2 \overline{BW} . h; \quad . \quad (1)$$

$$Br \sin\theta + W \sec\theta \cos\theta\, qr \sin\theta + W \sec\theta \sin\theta\, qr \cos\theta$$
$$= 4\frac{B}{g}\pi^2 N^2 r \sin\theta\, h; \quad (2)$$

$$B + qW(2 \sec\theta \cos\theta) = 4\frac{B}{g}\pi^2 N^2 h; \quad . \quad . \quad . \quad (3)$$

$$B + 2qW = 4\frac{B}{g}\pi^2 N^2 h; \quad . \quad . \quad . \quad (4)$$

and

$$h = g\frac{(B + 2qW)}{4B\pi^2 N^2}; \; N = \sqrt{\frac{(B + 2qW)g}{4B\pi^2 h}}. \quad . \quad . \quad (5)$$

In Porter's governor, $q = 1$, the balls being placed at D and F.

When $W = 0$, $N = \dfrac{g}{4\pi^2 h}$ as before, and we find that, while the common governor has a value of h, and a speed of rotation entirely independent of the weight or size of the balls; this is not true of the weighted governor.

For the common governor, we have, at any radius of rotation, y, and corresponding altitude h,

$$h : y :: B : B\frac{a^2y}{g};$$

a being the angular velocity; and, for the loaded governor,

$$h' : y' :: B + 2W : B\frac{a^2y}{g};$$

$$h = \frac{g}{a^2}; \quad h' = \frac{\left(1 + \frac{2W}{B}\right)g}{a^2};$$

$$\frac{h}{h'} = \frac{1}{1 + \frac{2W}{B}}; \quad \frac{\Delta h}{\Delta h'} = \frac{1}{1 + \frac{2W}{A}}; \quad . \quad . \quad . \quad (6)$$

as already shown by equation (5).

Thus the altitude of the simple pendulum is less than that of the loaded governor in this ratio. The altitude for the loaded governor is computed by multiplying that of the unloaded instrument by the quantity $1 + \frac{2W}{B}$, and the loaded governor is more sensitive in the same proportion. The office of the applied load is evidently to hold the balls down, while permitting higher speed of rotation and the storage of more energy.

In the case of loaded governors, as is seen, we find:

(1) Stability is increased by increasing the load with properly increased speed.

(2) Stability is gained by throwing the point of suspension of the arms out from the spindles.

(3) Stability increases with the range of movement of the load.

(4) With equal range, the stability increases with decreasing length of the suspending arms.

(5) The higher the ratio of load to weight of balls, the greater should be the angular velocity.

(6) The longer the suspending rods, the slower the speed of rotation.

The weight of suspending rods, usually of little effect in the unloaded governor, may have considerable importance where, as in many loaded governors, the balls are comparatively small and the speed great. The loss of sensitiveness of a governor is directly proportional to the resistance producing it, as the friction of the mechanism or a stuffing-box at the valve-chest.

98. The Isochronous or Astatic Governor, or that in which the position of equilibrium can only be preserved at one defined speed, is evidently an apparatus of which the distinguishing feature is its condition, at normal speed, of unstable equilibrium. Its general theory is the following:

In order that a permanent condition of equilibrium shall be secured, the work done upon the balls in the horizontal plane must be equal to that in the vertical plane, and

$$Xdx = Ydy; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

when X and Y measure the forces, and dx, dy the elementary spaces traversed by those forces during any small movement of the governor.

But Y may be taken in the fly-ball governor as the value of centrifugal force, and X as the vertical effort due to the weights of the balls, etc. Then, if we take a as the angular velocity, B as the weight of balls, etc., due to gravity, and y as the radius of the orbit of the centers of gravity of the balls,

$$Y = \frac{Ba^2y}{g}; \; X = B;$$

and

$$Bdx = B\frac{a^2y}{g}dy; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

whence

$$y^2 = 2\frac{g}{a^2}x; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

which is the equation of a parabola having its axis vertical, and of which the parameter is the constant quantity, $\frac{g}{a^2}$.

It thus follows that if it were possible to make a governor in which the path of the balls should be parabolic, such a governor would be, in principle, isochronous.

The same conclusion is at once deduced thus: the virtual height of this type of governor, the height from the plane of revolution of its balls to the centre of suspension, is the subnormal of the parabola; but the subnormal to the parabola is constant, and hence equilibrium can be maintained at but one speed, and will always exist at that speed whatever the position of the balls.

The parabolic governor, as such a governor is called, may be made either exact or approximate. It is made exact by compelling the balls to adhere precisely to the parabolic path; approximate by obtaining for them a line of vertical movement more or less closely, but not exactly, parabolic. Where the balls are suspended by a flexible line or strap, or by a spring, from a cheek-piece having for its outline, on the side in contact with the suspending element, the evolute of the parabola, an exact parabolic governor is obtained, as in Fig. 157; in which BM is the spindle, HL the cheek-piece, HE the "rod," E the ball, and $FK = AB = h$, the altitude of the governor.

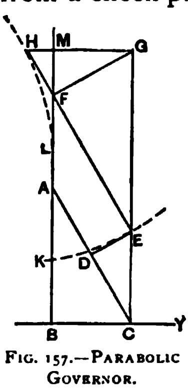

FIG. 157.—PARABOLIC GOVERNOR.

In this case,

$$MG = BC = y; \; EC = x; \; MH = -y'; \\ BM = GC = x';$$

and, with the origin at K, we shall have, when KE is taken as the proposed path of the balls,

$$y^2 = 2hx; \; x = \frac{y^2}{2h}.$$

Or, taking the origin at B, and making

$$\overline{AC}^2 = \overline{AB}^2 + \overline{BC}^2 = h^2 + y^2$$

$$\overline{AC} = \sqrt{h^2 + y^2}; \quad DC = \tfrac{1}{2}\sqrt{h^2 + y^2}. \quad . \quad . \quad . \quad (4)$$

$$\frac{CE}{DC} = \frac{AC}{AB} \quad \therefore \quad \frac{x}{\frac{1}{2}\sqrt{h^2 + y^2}} = \frac{\sqrt{h^2 + y^2}.}{h}$$

$$x = \tfrac{1}{2}\left(h + \frac{y^2}{h}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is the equation of the parabolic path of the governor-balls referred to the axes BM and BC, A being the focus and BY the directrix.

The use of this equation will enable the designer to lay off successive points in the curve, preparatory to designing the governor. AC being first drawn parallel to the proposed line of direction of the ball-rod, it is bisected at D, and DE and CE are drawn perpendicularly to AC and to BY, respectively; thus making E a point in the curve sought. EF is then made normal to this curve and parallel to AC. FG, parallel to DE and terminating in CG, locates a point G from which a line GH, parallel to BY and intersecting FE produced, determines a point H in the evolute HL. This is Rankine's geometrical construction.*

Algebraically, we have:

$$\overline{AC} = \overline{FE} = 2\overline{CD};$$

$$\overline{CE} : \overline{GE} :: \overline{DC} : \overline{FE} :: 1 : 2;$$

$$\overline{GE} = 2\overline{CE} = 2x;$$

* Machinery and Millwork, § 362.

But

$$\overline{GC} = \overline{GE} + \overline{CE} = \overline{MB} = x' = (2 + 1)x;$$

$$\therefore \quad x' = 3x. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

Also,

$$\overline{HM} : \overline{HF} :: \overline{BC} : \overline{AC} :: y' : \sqrt{h^2 + y^2};$$

$$\overline{HM} = \frac{\overline{HF} \times y}{\sqrt{h^2 + y^2}};$$

But

$$\overline{HF} = \frac{\overline{FG}^2}{\sqrt{h^2 + y^2}}; \quad \overline{FG} : FE :: y : h;$$

$$\overline{FE} = AC; \quad \therefore \quad \overline{FG}^2 = \frac{\overline{AC}^2 \times y^2}{h^2} = \frac{y^2(h^2 + y^2)}{h^2}$$

$$\overline{HF} = \frac{y^2(h^2 + y^2)}{h^2 \sqrt{h^2 + y^2}};$$

$$\overline{HM} = \frac{y^2(h^2 + y^2) \times y}{h^2(h^2 + y^2)} = \frac{y^3}{h^2} = y^1$$

and

$$-y^1 = \frac{y^3}{h^2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

and values of x and y thus obtained enable the designer to lay out the cheek HL.

The effort exerted vertically by the balls of the parabolic governor when speed changes is readily determined thus:

Let W be the collective weight of balls, N the number of revolutions per second, and ΔN the variation of speed, the radius of the orbit in which the balls constantly move being y. Then the vertical force is

$$X = W; \quad \text{and} \quad Y = \frac{W}{g} N^2 y;$$

$$X = Y \frac{g}{N^2 y}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

If the speed be increased to $N + \Delta N$,

$$X' = X + P = W + P,$$

P being the effort exerted to move the regulating mechanism, and

$$X' : X :: Y' : Y = \frac{W}{g} y(N + \Delta N)^2 : \frac{W}{g} N^2 y.$$

$$X' = W \frac{(N + \Delta N)^2}{N^2} = W\left(1 + \frac{2\Delta N}{N}\right);$$

since ΔN^2 may be neglected; and then we obtain

$$P = X' - X = W\left(1 + \frac{2\Delta N}{N}\right) - W = 2W\frac{\Delta N}{N}. \quad . \quad (9)$$

The power of the governor is thus seen to be solely dependent upon the speed and its fluctuations and on the weight of the balls.

The size and weight of ball for any known case can be computed when the resistance to motion of the regulating train and the allowable variation of speed are given, thus :

From (10),

$$W = \frac{PN}{2\Delta N}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If R be the resistance to be overcome, and s the distance through which it must act to correct the variation of speed ΔN; and if s' be the distance through which the balls must rise or fall in the same operation, we must have

$$Ws' = Ps, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and

$$W = P\frac{s}{s'} \cdot \frac{N}{2\Delta N}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The quantity $\frac{\Delta N}{N}$ is sometimes called the "coefficient of fluctuation."

99. **Approximate Parabolic Governors** are those in which the path of the balls is approximately, but not precisely, parabolic, and their action is thus only approximately isochronal. An approximate isochronism is often found more desirable than absolute perfection in this respect. Where the governor is in perfect balance, and free to move indefinitely at the slightest alteration of speed, sudden changes of load are apt to produce wide fluctuations of the adjusting mechanism, and thus to introduce "racing," or corresponding momentary, but yet serious, departures from the correct speed of engine; the inertia and energy of the moving parts of the governor continually throwing it from one side to the other of the mean position, and giving an effort alternately in excess of the resistance and the reverse. Where this adjustment is found desirable or necessary, this difficulty is usually overcome by the introduction of a "dash-pot" to control the motion of the governor.* The common fly-ball governor is often so controlled.

* A "dash-pot" is a cylinder having a loosely-fitted piston and filled with

and the shaft-governors of the Hartwell type, to be described, must often be thus fitted to secure perfect working.

In Farcot's governor, this approximate isochronism and an approach to the parabolic path of balls is obtained by hanging them from points of suspension on the opposite side of the spindle, the arms thus crossing at the centre-line of the latter; the centre of suspension being so chosen that the vertical orbit described by the latter shall as nearly as possible coincide with the parabolic in that part of the curve at which the intended working path ..lls. A light spring, resisting the rise of the balls, aids in securing sensitiveness.

The loaded approximate governor may be given an increased speed of rotation and greater power, as in the accompanying figure, in which a Steinlen governor is shown. The method of hanging and guiding the balls here shown gives still more perfect approximation to isochronism by causing a more rapid reduction of the altitude from the plane of the balls to the virtual point of suspension, as the balls separate.

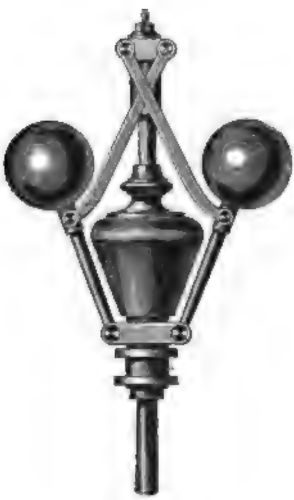

FIG. 158.—LOADED GOVERNOR.

Obviously no astatic governor is suited for use in connection with a detached train, like that of a water-wheel governor. Its settlement into any position that it may occupy when normal speed is reached is likely, at times, to leave it engaged with the train, thus producing irregularity. It must, at speed, always settle back to its normal place.

In the Babcock & Wilcox governor the balls, N, are hung upon arms in the usual manner, which arms are jointed at their

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oil or other fluid which will resist strongly all sudden motion while permitting easy movement at low velocities. See Rankine: Machinery and Millwork; § 364.

upper ends to a head attached to the rod, *o*, which slides within the hollow shaft that drives the balls; the motion being communicated through the radius-rods, *p*, which are jointed at their lower ends to the gearing-shaft, and at their upper ends to the centre of the arms, *n*. The rods, *p*, are half the length of the arms, *n*, measuring from the centre of the ball, and it will be readily seen that in consequence of this arrangement the arms, *n*, and rods, *p*, form a parallel motion and compel the balls to move outward in a horizontal plane.

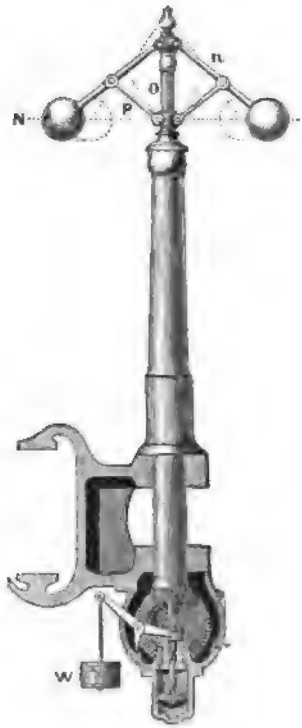

FIG. 159.—THE BABCOCK & WILCOX GOVERNOR.

In the ordinary pendulum governor the balls move in the arc of a circle and rise as they extend. It therefore requires an increased speed to maintain them in their advanced position. The engine must consequently run faster when the load is light than when it is heavy, and such is the case with all ordinary governors. In this improved governor it will be seen that the gravity of the balls has no tendency to move them in either direction, and exerts no influence whatever upon the speed of the engine. The centrifugal force causes them to diverge, and a weight, *W*, tends to bring them towards the shaft. When, therefore, these two forces are in equilibrium, the balls will remain in the same position, but as either preponderates they are moved in a corresponding manner, thus affecting the speed of the engine by varying the amount of cut-off. The weight, *W*, is supported upon a bent lever which is so proportioned that the centrifugal force, at any given speed, will just balance the weight in all positions. The speed of the engine will therefore remain at that fixed

point with all variations of load or pressure of steam; for any increase or diminution will cause either the balls or weight to preponderate and the point of cut-off to be changed until the speed is again brought to the standard where the two forces are in equilibrium.

Any desired speed can be obtained by altering the weight, *W*, and the action of the governor will be as perfect in one case as in any other. A spiral on the rod, *o*, serves to advance or retire the crank, *m*, relatively to the main crank, so as to cause the cut-off to occur earlier or later in the stroke, as the balls diverge or converge; and the amount of this adjustment is such that the cut-off may be varied as required.

A dash-pot at the lower end of the spindle prevents any "racing" of governor or of engine.

Bourne's governor had its balls arranged to slide horizontally on a rod secured at its middle-point to the spindle of the governor. A spring counterbalanced the balls, and the regulating action was like that of the Babcock & Wilcox governor. This apparatus was applied in the steamer San Juan as early as 1836, to adjust "throttle-valves" in the steam-pipes, and also in the injection-pipe supplying water to the condensers.*

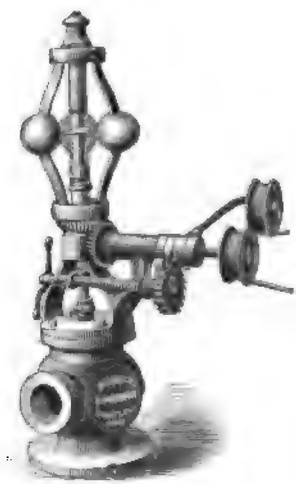

FIG. 160.—PICKERING GOVERNOR.

Silver's marine spring governor consisted of two crossed arms, each carrying balls at each extremity, balancing about a pin on the governor spindle. Centrifugal force was equilibrated by a helical spring coiled on the spindle.

In Pickering's governor, Fig. 160, the balls are carried each at the middle on flat springs curved in the form of a double cyma, and attached, at the lower extremity, to the driving-spindle, and, at the upper end, to a slide which is connected to the stem of a balanced valve.

* Bourne; Steam-engine.

100. **Spring Governors,** or those in which the action of a spring is substituted for that of gravity, are of many forms, of which probably the earliest was that of Foucault, who modified the action of the Watt governor, in the direction of isochronism, by the introduction of a steel spring offering an increasing resistance as the balls moved outward. The usual forms are, however, so arranged that the equilibration of the centrifugal effort of revolving masses is obtained solely by the action of a steel spring; the action of gravity being eliminated by the kinematic arrangement of the system.

It is evident that the relations of the variations of the two forces may be modified very greatly by the form given the mechanism transmitting the forces exerted between the springs and the balls of the governor. The simplest system would seem likely to be that in which the two opposing elements of the mechanical system are so connected as to be directly opposed to one another. It has been seen that the measure of centrifugal force is

$$F = \frac{W}{g} a^2 R; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

its magnitude being directly proportional to R, the radius of the circle in which the masses of weight, W, revolve, and to the square of the angular velocity, or speed of revolution. Experiment upon coiled (helical) springs, such as are very generally used in this class of governor, shows that their resistance to compression or to extension, P, varies in such manner that the momentary load is

$$P = pe; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

when p is a constant and e the variation of length observed, the resistance being proportional to the alteration of length of spring.

Were a direct connection made, as above suggested, we should have, and can practically make,

$$pR = W\frac{a^2 R}{g}; \quad p = W\frac{a^2}{g}; \quad . \quad . \quad . \quad . \quad . \quad (3)$$

provided the spring is unloaded when the balls are at the centre, or is loaded proportionally to their distance from the centre, and the quantity p being the load in pounds or in kilogrammes, which would make e unity in feet or in metres, whichever system of measures may be adopted for g.

It has been seen that the condition of isochronism is that, throughout the range of action of the governor,

$$Xdx = Ydy. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Here, if the radius of the orbit of the balls is R,

$$X = Cx; \quad Y = \frac{W}{g} a^2 R; \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and

$$dx = dy; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

conditions which are all at the same time consistent with the construction of this governor, and with the essential principles of isochronism.

For this class of governor generally, we have

$$p(R \pm a) = \frac{W}{g} a^2 R; \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where a represents the distance from the centre at which the spring is unloaded, or a constant having such a value as experiment may prove necessary to make the equation true. For isochronous governors, $a = 0$, and p is the load which will stretch the spring a distance unity.

Nearly all governors of this class are mounted on the engine-shaft, and are called "shaft-governors."

The general arrangement of parts in this type, from the pioneer form of Hartnell and Guthrie to the latest, is substantially the same. An admirable illustration of the class is here given in the figure showing that designed by Mr. Westing-

house; in which the weights are arranged symmetrically about the shaft, and are held in equilibrium with centrifugal force by helical springs attached to the governor case; which, in this

FIG. 161.—THE SHAFT GOVERNOR.

instance, is closed and oil-tight, and serves as a tank for the lubricant, and thus insures free and constant lubrication.

It will be observed that the Babcock and Wilcox, the Buckeye, Ball, Ide, and other forms of governor elsewhere described, have dash-pots attached to prevent oscillation when at work and consequent "racing" of the engine. The dash-pot is attached to all forms of governor, and in a great variety of forms; but it is not so useful in perfectly astatic governors. A certain amount of stability must exist to permit successful operation of the dash-pot.*

101. The Theory of the Coiled Spring is the following:

Assume, in order to secure perfect action and a constant ratio of load-stress to variation of length, that the spring is an

* Wischnegradski; Civil-Ingenieur, 1877.

absolutely true helix, and that it is always loaded by tension rather than compression. Were the spring an imperfect helix, one section would assume a different rate of variation from another, or even take a set; were it compressed, instead of extended, by its load, it might yield laterally.

Assume that the line of pull is precisely along the axis of the helix; were it not thus central, it would tend to distort the spring. To secure a correct location of the line of action of the applied and resisting forces, the spring is usually either terminated by leading its end along the axis, or is set in properly fitted caps; the former method is common in spring-balances, the latter in the springs of the steam-engine indicator.

Assume the section of the wire of which the spring is composed, as well as that of the spring itself, to be circular, this form being practically best and the theory of its action and the laws involved being better established than for other forms.

Let r_1 = the radius of the helix;
d_1 = the diameter of the helix;
n = number of coils;
d' = diameter of the wire;
r' = radius of the wire;
C = the coefficient of transverse elasticity of the metal;
W, W_1 = the loads;
e, e_1 = the corresponding extension when W is any, and W_1 is the maximum permissible, load.

The twisting moment acting on the wire of the spring, and the moment of resistance to torsion so called out, are the two equal quantities causing equilibrium at any instant.

Let M = this moment;
θ = the angle of torsion in circular measure;
x = length of wire loaded;
q = actual shearing stress;
f = safe " " .

Then, since, within the elastic limit, the stress varies as the angle of torsion, inversely as the length, and directly as the

values of C and of the radius at which any element is assumed to act, the shearing stress is

$$q = Cr\frac{d\theta}{dx}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The area of any one of the elementary rings into which the section of the wire may be conceived to be divided is

$$a = 2\pi r dr;$$

when r' = the external radius of the wire, and f is the maximum allowable stress,

$$q = f\frac{r}{r'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For each element the moment is

$$m = 2\pi r dr \,.\, \frac{fr}{r'} \,.\, r.$$

$$M = \frac{2\pi f}{r'}\int_0^{r'} r^3 dr = \tfrac{1}{2}\pi f r'^3 = 1.5708 f r'^3. \quad . \quad . \quad (3)$$

But

$$\frac{d\theta}{dx} = \frac{q}{Cr};$$

and, since the wire is of uniform texture and quality,

$$\frac{d\theta}{dx} = \frac{\theta}{x} \quad \text{and} \quad \theta = \frac{xq}{Cr};$$

and since $\frac{q}{r} = \frac{f}{r'}$,

$$\theta = \frac{fx}{Cr'} = \frac{2fx}{Cd};$$

$$\frac{M}{\theta} = \tfrac{1}{2}\pi f r'^3 \div \frac{fx}{Cr'} = \frac{C\pi r'^4}{2x} = \tfrac{1}{32}\pi C\frac{d^4}{x}. \quad . \quad . \quad (4)$$

In the case of the helical spring,

$$M = Wr_1; \quad \theta = \frac{e}{r_1}; \quad x = 2\pi r_1 n;$$

and

$$\frac{Wr'r_1}{e} = \frac{\pi}{32} \cdot \frac{Cd'^4}{2\pi r_1 n};$$

and the elasticity is measured by

$$E = \frac{W}{e} = \frac{Cd'^4}{64nr_1^3}; \quad e = \frac{W}{E}; \quad W = \frac{E}{e}, \quad . \quad . \quad . \quad (5)$$

The maximum load, W_1, would be obtained thus:

$$M = \tfrac{1}{16}\pi f d^3 = W_1 r_1; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$W_1 = \frac{\pi f d^3}{16 r_1} = 0.196\frac{fd^3}{r_1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and

$$E_1 = \frac{W_1}{e_1} = \frac{0.196 fd^3}{r_1 e_1} = \frac{Cd^4}{64nr_1^3} = \frac{W}{e}.$$

$$\frac{W_1}{E_1} = e_1 = 12.6\frac{fnr_1^2}{Cd}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The value of C may be taken at about 10,000,000 pounds on the square inch, or about 705,000 kilogrammetres on the square centimetre. The value of f should be made not far from 0.003 these figures or 30,000 to 35,000 in British and 2100 to 2500 in metric measures. The British Board of Trade rule allows, as a maximum, for safety-valve construction $d = \sqrt[3]{(Wd_1 \div c)}$; c being taken at 8000 for round and 11,000 for square wire.

102. The Theory of Spring Governors includes the principles controlling the action of the spring and its relation to that force, centrifugal or other, which balances its effort. In the ordinary spring governors, such as have been described, a mass revolves about an axis so connected with the motor to

be regulated as to move at a rate precisely proportional to the speed of the latter. The centrifugal force thus generated is equilibrated by the action of a spring so adjusted as to produce a balance at the desired speed. In the astatic or isochronous form, this balance can only occur at the correct speed, but at that speed it is maintained in all positions of the governor. In the static, or non-isochronous governor, this balance is obtained in one position, and in one only, at any one speed; it may, at other speeds, rest in other positions similarly definite for each speed.

It is obvious that, should the resistance of the spring vary exactly as centrifugal force at constant speed with varying radius, the governor must be isochronous; but if the spring increase in resistance faster than the centrifugal force increases, the governor will be in static equilibrium and non-isochronous. Should the resistance of the spring increase less rapidly, equilibrium is at once lost, and the apparatus is no longer effective or a governor. By giving such initial excess of tension as to produce a more rapid increase of spring resistance, that of centrifugal force, any desired stability of governor can be secured, at the sacrifice, however, of sensitiveness and—inertia apart—of precision, of regulation.

Thus the two conditions which determine the proportions of this governor are the magnitude of the initial tension on the spring and the method of variation of its resistance. For an isochronous governor, the moment of the initial tension about the pivot must be just equal to the moment of the effort of the centrifugal force of the governor-ball sustained by that spring, when the engine is at speed and the ball at the inner extremity of its total range; and, further, that moment must vary proportionally with the radius, R.

We may now proportion such a governor, taking, first, the simplest case, that in which the spring and ball have equal lever arms, and $W = F$. We have for F,

$$F = \frac{4\pi^2 R N^2 W_1}{g},$$

which becomes, when $g = 32.166$ feet, N is the number of revolutions per minute, W is in pounds, and R in feet,

$$F = 0.000,341\, RN^2 W^1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

For R in *inches*,

$$F = 0.000,028\, RN^2 W^1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Assuming the spring to be helical and in tension, and calling the pull on it P, we shall have, as a maximum,

$$P_{max} = F_{max} = 0.000,028\, R_{max}\, N^2 W^1 = W_1 = 0.196 \frac{fd^3}{r_1} . \quad (3)$$

and taking $f = 30,000$ for British measure, and making the spring of common proportions, such that $d = \frac{1}{2} r_1$,

$$W_1 = P = 368 r_1^2 = 0.000,028\, R_1 N^2 W;$$

$$r_1 = \sqrt[3]{0.0,000,000,075\, R_1 N^2 W^1}; \quad . \quad . \quad . \quad . \quad (4)$$

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and the diameter of wire, d_1 required when R_1 and N are given by measurement at the engine, and W^1 by the assumption, or the measurement, of the work to be done by the governor, is

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$$d_1 = \sqrt[3]{0.00,000,006\, R_1 N^2 W^1} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

If the size of a single ball and its connections are such that W^1 may be taken, as their equivalent, at 10 pounds, and if the maximum radius R^1 be one foot and the revolutions 200 per minute,

$$d_1 = \sqrt[3]{0.00,000,006 \times 12 \times 40,000 + 10} = 0.65'', \text{ nearly.}$$

When, as is common, the spring is so attached that, while exerting the same moment, it has a longer or shorter lever-

arm, it must be made of correspondingly smaller or larger wire, as the case may demand. In such cases, the value of W' becomes

$$W' = \frac{W''l''}{l'}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

in which expression W'' is the effort of the spring and l_1 and l'' are the lever arms of the ball and the spring respectively.

For the example just taken, assuming the spring attached half way between the ball and the point about which the ball-arm swings $\frac{l'}{l''} = 2$; $W' = 2\ W''$, the equivalent pull at the ball being one-half that actually exerted by the spring, the wire of which has a diameter

$$d_1 = \sqrt[3]{0.00,000,006 \times 12 \times 40,000 \times 20} = 0.75''.$$

For 300 revolutions, direct action,

$$d_1 = 0.75'' \text{ nearly};$$

a very common size.

The size of spring and wire having been thus determined, the length of the coil is obtained by ascertaining the range through which the spring must act. This range is that which it would traverse were it so attached, and the ball so adjusted on its arm, that the latter could swing in quite to the axis about which the governor revolves; at which point its centrifugal force would vanish and the tension on the spring would become zero. This range is measured by e_1 (§ 101) and

$$e_1 = \frac{12.6nfr_1^2}{Cd} = R_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

Taking $C = 10,000,000$, $d = \frac{1}{2} r_1$, and $f = 30,000$, the number of coils is,

$$n = \frac{13.2R_1}{r_1}, \text{ nearly} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

If we take $R_1 = 12$; $r_1 = 0.6$; $\frac{R_1}{r_1} = 20$; $n = 26$ nearly; and the spring has a minimum length,

$$L = nd_1 = 2nr_1 = 31.2''.$$

Such length of spring is rarely used or possible, and one-half this length and extent are more usual. Where isochronism is desired to be fully secured, special devices of arrangement of the arms and levers are adopted, permitting more or less accurate adjustment of initial tension and of stresses as required.

For any intermediate position of the governor, the value of two equilibrating forces is known to be

$$W = P = \frac{Cd^4e}{64nr_1^3}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which expression all the quantities in the left-hand member are known.

Taking $C = 10,000,000$, $d = \frac{1}{2}r_1$, and $n = L \div d_1$,

$$W = P = 2,500 \frac{Rd^3}{L}, \text{ nearly} \quad . \quad . \quad . \quad . \quad . \quad (10)$$

in pounds.

The variations in quality of the steels used for springs, and especially the variations of temper, are so great in practice, that it is never safe to rely entirely upon computation for the final dimensioning and setting of this class of governors. They may be thus given approximately correct proportions, and finally the sizes and adjustments exactly determined by trial.

This type of governor, when very nicely adjusted, is liable to fluctuations due to inertia of parts. The defect is remedied, as already noted, by the use of a dash-pot.

The plan adopted by Mr. Ball is the following:—The springs, *D*, are given full theoretic tension by a stretch corresponding to the distance from the centre of weight to centre of shaft. The spring, *S*, attached to the weight arm, has no

initial tension, and, acting in combination with the other springs, produces the effect of springs with less than full theoretic tension, thus giving stability due to such adjustment, and the same variation of speed, provided the dash-pot be dispensed with.

The combination of this dash-pot with a supplemental spring makes it possible to produce steady working. This dash-pot consists of a cylinder filled with oil, having a piston with a small aperture through which the fluid may pass as the

FIG. 162.—BALL GOVERNOR AND DASH-POT.

piston moves. The piston-rod of the dash-pot is attached to the supplemental spring, which arrangement provides a yielding base. When motion of the governor-weights takes place in either direction, this spring is put under tension for the moment, and gives stability to the governor; but immediately the action of the dash-pot releases the tension, the spring returns to its

normal condition, in which it is not a factor in the speed of the engine, which is determined entirely by the main springs, and, as these springs have full theoretic tension, the engine must assume the same rate of speed at every point of cut-off, without the instability often observed with full theoretic tension.

The function of the dash-pot is ordinarily to resist intermittent disturbing influences, and to counteract the disturbing action of unbalanced parts.

The action of the supplemental spring tends to produce freedom of motion of the governing mechanism without preventing prompt motion.

A similar arrangement is seen in the Ide governor, Fig. 163,

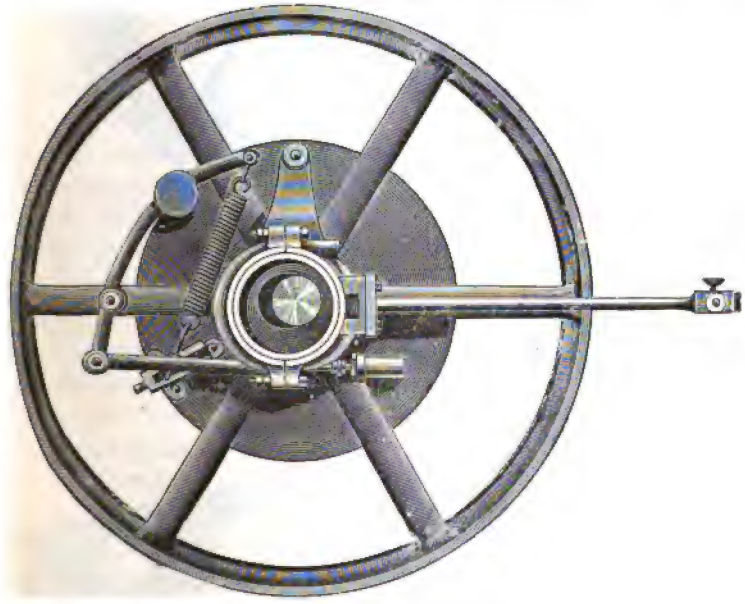


FIG. 163.—THE IDE GOVERNOR.

A dash-pot attached to the end of one of the levers of the governor controls the movement of the weights, preventing any sudden movement when a great change of load occurs suddenly, and by its use a more sensitive adjustment of the springs and weights can be made, and a closer and more perfect regulation of speed can be obtained.

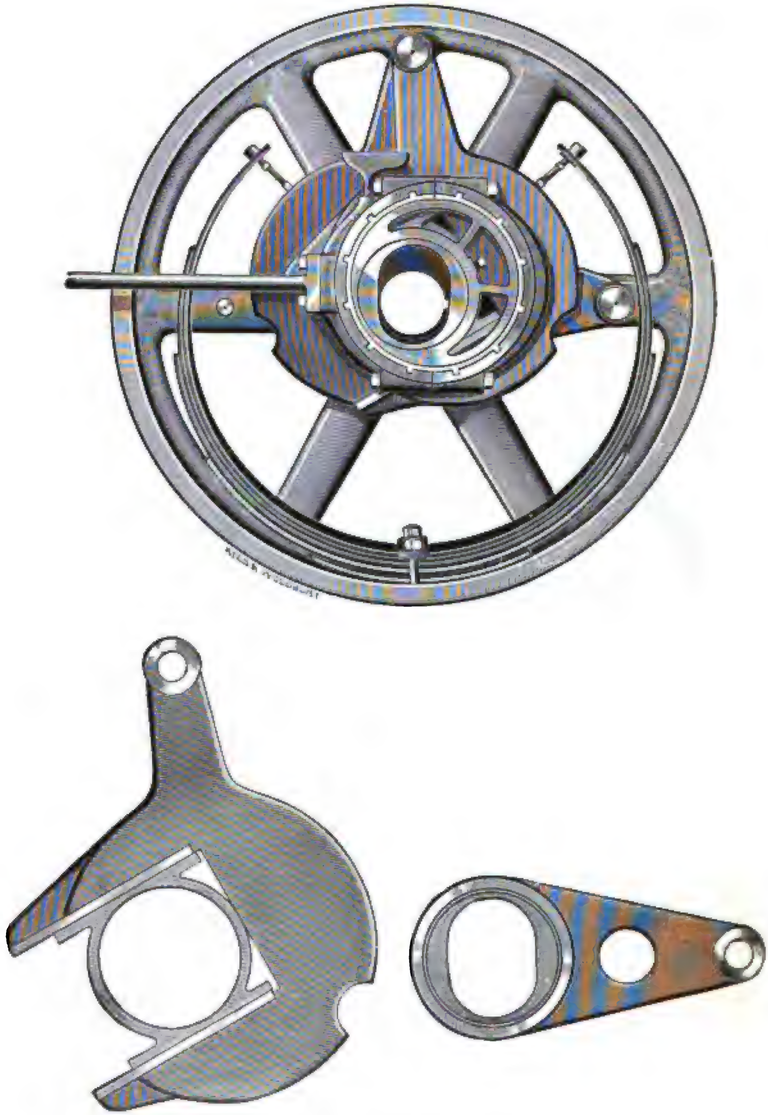


FIG. 164.—GOVERNOR AND DETAILS.

The dash-pot is here directly attached and pulls in line with the lever with which it is connected; no spring being interposed.

The shaft governor is a "safety governor," and if any part breaks or becomes deranged in any way, the result is to stop the engine. The use of the dash-pot was probably resorted to at an early date for the purpose above described.

Messrs. McIntosh and Seymour have also devised an interesting modification of the Hartnell type of governor. This is shown in Fig. 164. It consists of a pair of pivoted weights, one of which is shown separately at *A*, having inclined jaws, in which slide two blocks. These blocks turn freely on a boss upon the pendulum, shown at *B*, to which the eccentric is attached, and which is free to swing across the shaft. The pendulum is pivoted in such a way that while the cut-off changes from five-eighths to zero, the steam-lead does not vary. The inclination of the jaws in the weights is such that, through the action of friction, their position is not influenced by the action of the valve and rod, and yet they are always in statical equilibrium, and have no tendency to race. The spring bears upon the weights through the hardened steel pins, one end of each pin resting in a hardened cup on the end of the spring and the other in a slot in the weight. These slots are made deep, so that the pin bears upon the centre of gravity of the weight, and the pressure of the spring directly opposes the centrifugal force.

The arrangement of the governor on the engine is seen in the accompanying full-page engraving of the tandem compound of this make.

The engine itself illustrates a now standard construction. The high-pressure cylinder and the receiver are jacketed. The two valves are placed on opposite sides of the engine to secure direct connection and accessibility. The receiver takes jacket-steam from the high-pressure cylinder jacket and returns all water of condensation to the boiler. The governor system actuates the high-pressure valve, the low-pressure eccentric being fixed. The general proportions of parts can be judged from the illustration.

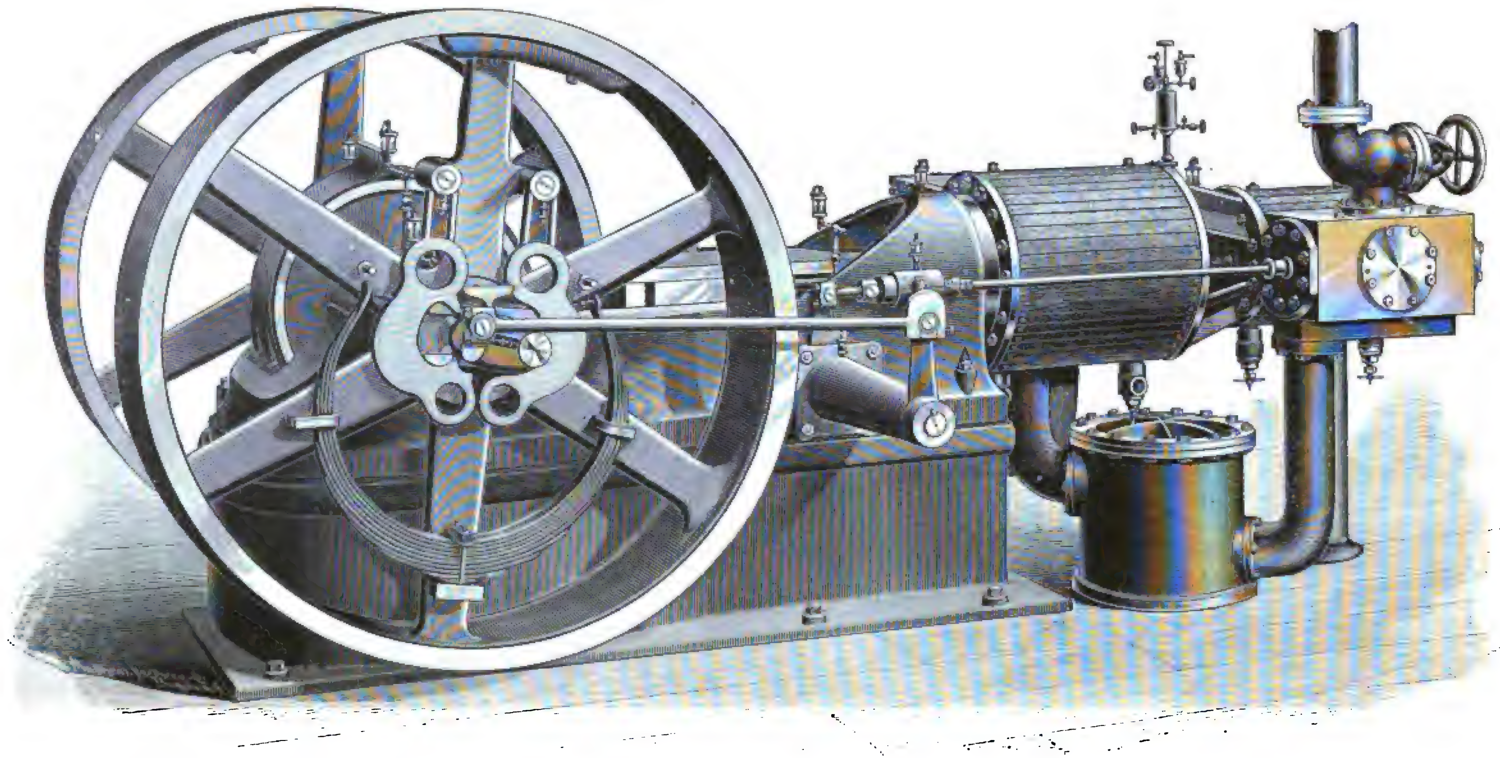


Fig. 165.—The McIntosh and Seymour Compound Engine.

103. The Differential Governor consists of a train of mechanism, of which a part is driven by the engine and at a speed proportional to that of the latter; while another element is regulated in some manner by a governor or revolving pendulum of proper form and method of attachment, the relative motion of the two being utilized, whenever a variation of engine-speed takes place, in the adjustment of the steam-supply. An example of this type is seen in Siemens' governor, in the accompanying illustration, as given by Rankine.

A is a vertical spindle; *C* is a pulley driven by the engine and fixed to a bevel-wheel, as seen below; *E* is a wheel having the same pitch-cone, and to this wheel are hung the masses *B*, which form sectors of a ring, and are surrounded by a casing *F*. They are adjusted so as nearly to touch this casing; should they fly outwards and touch the casing, they are retarded by the friction. *G*, *G* are horizontal arms capable of rotation about *A*, and carrying bevel-wheels, which rest on *E*, support *C*, and transmit motion from *C* to *E*. *H* projects, and has a rod attached to its extremity to move the valve.

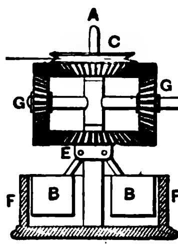


FIG. 166.—SIEMENS GOVERNOR.

When *C* rotates with an angular velocity equal and contrary to *E*, the arms *G*, *G* remain at rest; otherwise they move in such manner as properly to adjust the valve.

104. The best Disengagement-governor is that used for water-wheels, and rarely for the steam-engine. *AA* is the spindle; *B*, a slider, hung from the ball-rods by links *C*, *C*. *D* is a cam projecting from the slider.

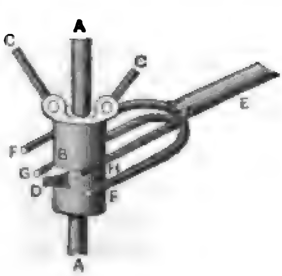


FIG. 167.—DISENGAGEMENT-GOVERNOR.

E is a lever turning about a vertical axis, and provided with a fork, *F*, *F*, *G*, *H*. *F*, *F* are far enough apart to clear *D*, when the spindle is turning at its proper speeds, and the ball-rods and slider in the middle position; the lever *E* is in its middle position also. *G* is below, and *H* above, the level of *F*, *F*; when the lever is in its

middle position, the distance of *G* and *H* from the cylindrical surface of the slider, *B* is one-half of the distance of *F*, *F*. When the spindle falls below its proper speed, the slider moves until *D* strikes *G*, and drives the lever *E* to one side. Should the spindle turn faster than the proper speed, the slider rises until the tooth *D* strikes *H*, and drives *E* to the other side. *E* acts on the clutch of a reversing-gear, and moves the regulating device, by a train either of gearing or of belted pulleys.

105. **Fluid Governors** employ various kinds of fluids and act in equally various ways. In those in which water is employed, the instrument either involves the action of the common plunger-pump, or is an application of the force of inertia of fluid masses to the movement of mechanism. When air is the fluid, the governor is either a system of fans, or involves the use of a pump or a bellows. An example of the first class is seen in one of the earliest of the governors, Pitcher's "hydraulic regulator," in which two small pumps are set in a reservoir and surrounded by water. One of the pump-pistons is driven by the engine to be regulated, at such speed as to force a determinate volume of water into the chamber of the other pump, and beneath the piston of the latter. Both pistons being solid, and of the same size, the second would exactly reproduce the motion of the first, assuming no flow of water to occur into or out of the system. But a flow being permitted through the suction valves of the first, and an opening being arranged for discharge from the second, the motion of the latter is greater or less as its eduction orifice is more or less closed. When the engine moves too rapidly, the second piston rises and falls within a higher range; when too slowly, this range is lower; at normal speed it does not act on the throttle-valve; but at too high or too low speed its chocks move the valve, closing it or opening it as required.

The Huntoon governor illustrates the application of a very different principle. In this case a fan revolves within a closed casing partly filled with water or oil. At high speeds great resistance is met and is utilized in the moving of a counter-balanced train closing the throttle-valve; at lower speed than

the normal, the valve is opened by the counter-balance, which is then able to more than equilibrate the action of the fan. These governors are, in principle, isochronous.

A bellows governor has been sometimes employed, in which air is the working fluid, and bellows take the place of pumps in an arrangement somewhat similar in principle to the first of the above-described devices. The fan-governor is one in which the resistance to the motion of fans revolving in the air is utilized much as is that of the second of those forms of governor.

All these governors may obviously be classed as isochronous governors; since they endeavor to do their work whenever off speed and persist indefinitely.

106. Marine Governors, having an essentially different function from all other classes of regulator, are commonly constructed on equally different principles. The work of the marine engine, in smooth water, is constant at any given speed, and, the steam-supply being kept uniform, it will preserve precisely the same rate of motion indefinitely. The marine engine, under such circumstances, thus requires no governor; the valves and cut-off gear once set, no further care is demanded to preserve uniform speed than to see that the engine is properly lubricated and that it has an ample supply of steam at the right pressure.

In a sea-way, however, the pitching and rolling of the ship are often productive of sudden and extreme fluctuations of speed. At one moment the propeller is thrown nearly or quite out of water, and, resistance being diminished enormously, the engine instantly starts off at a velocity which is often both frightful and dangerous; the next instant, the wheel deeply buried in the sea, the speed as suddenly falls to a minimum. The higher speeds thus entail risk of accident; the lower produce ineffective and inadequate expenditure of power. A governor which shall prevent such violent and objectionable variations of engine-speed must be capable of acting, at such times, instantly and efficiently, and must permit the engine to move at least approximately at the

proposed and regular speed under all conditions of wind and sea.

It would be impracticable to so adjust any of the usual forms of governor as to secure safety against such extraordinary accelerations and retardations of speed, as the inertia of their heavy parts is found to be a serious obstacle to proportional variations of their own speed. It is in certain other forms, however, found practicable to take advantage of this very resistance to variation of speed of heavy masses to secure that action which governors are ordinarily unfitted to produce. In the most common type of marine governor, as originally introduced by Mr. Silver, the train of driving mechanism acts upon the spindle of the governor through a system of springs, which, being partially flexed in regular motion, their effort to restore themselves is met and equilibrated by a set of fans or vanes attached to a fly-wheel which constitutes the revolving mass giving the required inertia. A sudden acceleration of speed of the engine applies to those springs a force which is the greater as the acceleration is the greater, tending to correspondingly increase the speed of the governor; this effort being resisted by the inertia of the governor fly-wheel, the springs are compressed, and the relative motion thus produced between it and the axis is made the means of closing a throttle-valve. When the engine suddenly slows down, the reverse action occurs, the springs being relaxed, and the steam is supplied more freely by the opening of the valve. Approaching the regular and prescribed speed, the resistance of the vanes becomes once more equal to the effort of the partially flexed springs, and the valve is held open sufficiently to supply the needed quantity of steam to keep the engine at speed; thus both avoiding "racing" on the one hand and securing uniform speed on the other, by this simple mechanism.

The work of the marine governor on large engines is so heavy, usually, that an auxiliary device, as described in the next article, is commonly considered essential to its successful working.

107. **Power and Efficiency** are the two essential forms of

the measure of the value of the governor. The efficiency of the governor is a function both of its own effectiveness and of the character of the regulating apparatus to which it is to be attached. A governor, even in the least exacting work, should be efficient; it should be quick to feel a change of speed, powerful enough in action to overcome instantly all resistance, and free from internal resistances as well as unimpeded externally. The measure of the efficiency is found in the quickness, power, and certainty of its action, as well as in the nicety with which it comes to speed, finally, after having been disturbed. These qualities should be obtainable without sacrifice of lightness, compactness, or of freedom from danger of accident or of heating journals.

Specially powerful governors are often required to handle heavy valve-gearing where, at the same time, efficiency is also demanded. In such cases, the governor, instead of being directly connected, is sometimes arranged to adjust the valve of an accessory, or auxiliary, motor, as a small steam or hydraulic cylinder, which, in turn, moves the regulating apparatus. This is done, for example, with Murdoch's marine governor, and in some of the heavier classes of pumping and winding engines.

The Author has proposed, for special cases, in which prompt and sure control is demanded, to use a governor affecting the exhaust and producing cushion or compression-variation, as both an effective and an economical system.*

Since the proper ratio of expansion is determined mainly by the steam-pressure, and since any variation from that point is usually productive of reduction of efficiency, that ratio should, if practicable, be fixed at the best proportion for the steam-pressure adopted, and never changed. The adjustment of a throttle-valve by the governor is inadmissible, as it involves variation of pressure in the steam-chest and reduced efficiency; the steam and expansion lines should be permanently fixed for all loads. To throttle the exhaust by the action of the governor would undoubtedly give a means of regu-

* Trans. Am. Society Mech. Engrs.; June, 1881.

lation, but a costly one. It becomes evident that such a system of regulation must affect the back-pressure or the cushion-line.

It is obvious that the only other plan is variation of the power of the engine by alteration of the compression line. This is done where the Stephenson link is adopted. When the link is down, the ratio of expansion is determined by the lap and lead, and is usually not higher than $\frac{1}{3}$; as the link is raised, this ratio is increased, and with this change of the steam line occurs a simultaneous alteration of the point of release and of closure of the exhaust passages, resulting in increased compression.

The steam-engine may, therefore, be worked with a fixed cut-off, so attaching the governor as to determine the point of closing of the exhaust-valve instead of attaching the cut-off mechanism to the steam-valves. Properly constructed relief-valves should be employed to prevent danger from the influx of water with the steam.

Maximum economy of steam may perhaps be obtained by an expansion-gear in which, as in the locomotive valve-gear, increased expansion is accompanied by increased compression, but without that serious throttling along the steam line which usually characterizes the distribution by the link-motion.

The best form of valve-gear should be, if judged from the standpoint here taken, that which, combining variable expansion with variable compression, is capable of prompt and exact adjustment by a sensitive and efficient governor.

108. Safety and other Attachments to governors may complicate their construction and increase their cost to a notable extent; but some classes of governor should always be provided with a device which will insure safety. Where the governor is driven by a belt, it should always be, if possible, fitted with some arrangement by means of which the steam will be instantly shut off in case of breakage or slip of the belt.

Whatever the form of this device, it is vitally important that it should be certain and very prompt in its operation, especially on an engine working with considerable expansion;

since a change of the steam-distribution, permitting the engine to "follow full-stroke," is likely in such cases to produce dangerous acceleration of speed very quickly, and possibly in a single revolution. Many serious accidents have resulted from such action, producing destruction of the fly-wheel and even of the whole engine and its surroundings.

109. **The Choice of a Governor** is determined mainly by the considerations detailed in the preceding paragraphs, and especially in the second article in this chapter. The cost of the governor, either to the maker or the user, is a matter of too little importance, as a rule, to be permitted any weight practically in this choice. The best governor for any engine is that which will most certainly and constantly preserve the correct speed, and its selection is usually largely a question of adaptation to the special form of engine to which it is to be attached.

110. **The Fly-wheel or Balance-wheel** is employed on all except marine engines, and a few forms of rolling-mill engines; it has even been occasionally introduced on ship-board, with single engines, to give regularity of motion of rotation.

Its office has been seen to be that of a reservoir for storage of energy. When the speed of engine increases, the acceleration stores a quantity of energy measured by the sum of the products of the weights of the several parts so affected into the difference produced in the squares of their velocities:

$$\Delta E = \Sigma W \frac{(v_2^2 - v_1^2)}{2g}. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The energy thus stored is restored in the effort to retard any reduction of speed occurring later.

The effectiveness of the wheel is seen to increase as the weight of wheel and the square of the velocity increase, without limit. In the design of the wheel, the weight of rim is commonly made sufficient to give the needed smoothness of motion, and no attention is paid to weight of arms and other

parts, which are proportioned purely with a view to securing ample strength.

If v_1 be the mean velocity of the centre of the section of the rim, v_2 and v_3 its greatest and least velocities, the variation of energy is, for the weight W of rim,

$$\Delta E = W\frac{v_2^2 - v_3^2}{2g}. \qquad (2)$$

The mean energy would be

$$E_1 = W\frac{v_1^2}{2g}, \qquad (3)$$

and the ratio of the two is

$$\frac{\Delta E}{E} = \frac{v_2^2 - v_3^2}{v_1^2}.$$

and since $v_1 = \frac{1}{2}(v_2 + v_3)$, we have

$$\frac{\Delta E}{E_1} = \frac{v_2^2 - v_3^2}{v_1^2} = \frac{4(v_2^2 - v_3^2)}{(v_2 + v_3)^2} = \frac{4(v_2 - v_3)}{v_2 + v_3}.$$

and

$$\frac{\Delta E}{2E_1} = \frac{v_2 - v_3}{\frac{1}{2}(v_2 + v_3)} = \frac{v_2 - v_3}{v_1},$$

or

$$\frac{v_2 - v_3}{v_1} = \frac{1}{m} = \frac{g.\Delta E}{Wv_1^2}; \qquad (4)$$

which ratio is called by Rankine* the "coefficient of fluctuation of speed or of unsteadiness." The value of m varies enormously, with different classes of machinery, in even the best practice. It should usually exceed 50 in flour and cotton mills and in electric lighting; and it often becomes 100 for very fine work. In ordinary machine work, $m = 33$ to 50, and

* Machinery and Millwork, § 357.

in coarse and heavy work its value may fall still lower. For power hammers it drops even to 5. The higher the speed of rotation of engine, the smaller and the less costly the wheel, and, as a rule, the less the waste of power by friction on the shaft-journals.

According to Weisbach, the coefficient of fluctuation of speed has the following maximum allowable values:*

For hammer-trains	$\mu = \frac{1}{m} = 0.200$	
" pumps and gang-saws	0.050 to 0.033	
" grist-mills	0.040 " 0.030	
" looms and paper-mills	0.033 " 0.025	
" spinning-machinery	0.025 " 0.010	

In steam-engine construction the same authority adopts the value 0.030 for proportioning fly-wheels. Recent improvements in regulation have now made it practicable to construct engines in which the mean speed of revolution shall not be departed from, one per cent, and this being made a standard requirement, in some cases, as in electric lighting, involves the necessity of increasing the efficiency of balancing and of the fly-wheel to bring the fluctuation of the speed of rotation of the shaft within the same limit, revolution by revolution.

Thus, to maintain such uniform speed with a uniform load that its fluctuation shall not vary more than a given fraction, $\frac{1}{m}$, from the mean, the ratio of energy stored in the fly-wheel at mean speed to the energy supplied by the engine in a revolution must be not less than one half the quotient of the coefficient of fluctuation of this energy, divided by the coefficient of fluctuation of speed.

111. **The Required Weight of Wheel-rim** is, from the preceding,

$$W = \frac{mg \cdot \Delta E}{v_1^2} . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

*Vol. III., § 191.

Since, if a is the angular velocity of the wheel, and if I is the moment of inertia of the rim,

$$E_1 = \frac{a^2 I}{2g},$$

the required moment of inertia becomes

$$I = \frac{mg\Delta E}{a^2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and if r is the radius of the wheel,

$$W = \frac{mg\Delta E}{a^2 r^2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which expressions the values of mg have been seen to range from 80 feet to above 3200; the last figure being that of the very best cases in current practice.

For the steam-engine, taking

$$\Delta E = \frac{Wa^2r^2}{mg} = \frac{Wv_1^2}{mg};$$

and since

$$\Delta E = Wr \cdot \frac{v_2^2 - v_1^2}{2g} \propto \int pds \propto H.P.;$$

$$W = Cmg\frac{H.P.}{N^2D^2} = C'\frac{ASp}{N^2D^2}, \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which C is a constant, p the pressure on the piston, $H.P.$ the power of the engine, and N the revolutions per minute, and where D is the outside diameter of the wheel in feet, A, S, and p are the area of piston in square inches, its stroke in feet, and the steam-pressure in pounds per square inch. The Author

has, in designing engines with automatic valve-gear, used this last expression; thus,

$$W = 250{,}000\frac{ASp}{N^2D^2}.$$

Experience has also shown that we may usually safely take

$$W = \frac{aAS}{N^2D^2},$$

in which a ranges from ten to fifteen millions, according to character of work, averaging in ordinary practice about 12,000,000.

This gives a weight of wheel which answers well for the ordinary forms of non-condensing engine, as commonly operated with a ratio of expansion between 3 and 5.

In the exact calculation of size of wheel, it is obvious that we must first ascertain the nature, location, and amount of the maximum variations of energy to be counterbalanced by it. These, as already seen, may consist of variations of the applied effort on the piston and turning moment at the crank; they may be variations of the resistance due to changes of load; or they may be, and generally are, the resultant of both methods of disturbance of the balance between effort and resistance.

Commonly, the variations at the engine are so great that, if the wheel is designed properly to meet them and to reduce the coefficient of fluctuation to its required value, the engine will prove successful. In exceptional cases, as that of the rolling-mill engine, or special machines where the load is thrown on and off, at times, the variation on that side, or on both, must be allowed for, and ΔE obtained, as a total difference between the varying power and load.

The values of $\dfrac{\Delta E}{\int pds}$, the ratio of this variation of impelling energy to the total work of the engine for a single revolution,

was first determined, for usual cases, by General Morin, and given as follows:

VARIATIONS OF ENERGY, $\frac{\Delta E}{\int pds}$.

For $r = 1$.

$\frac{\text{Length rod}}{\text{Crank}}$	4	5	6	8
$\frac{\Delta E}{\int pds}$	0.132	.125	.118	.105

NON-CONDENSING ENGINES.

r variable; rod length = 5 *times crank.*

r	2	3	4	5
$\frac{\Delta E}{\int pds}$	.160	.186	.209	.232

CONDENSING ENGINES, AS ABOVE.

r	3	4	5	6	7	8
$\frac{\Delta E}{\int pds}$	.163	.173	.178	.184	.189	.191

The value of this ratio is thus between 0.15 and 0.20 for all the usual forms of engine. It may be safely taken at the higher figure in designing mill-engines, and the value of ΔE is to be, in that case, assumed at $0.20U$, where U is the work done in a single revolution. The weight of wheel-rim, in pounds, then becomes

$$W = 0.2\frac{mgU}{v_1^2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and taking m at 100 for close regulation

$$W = 640\frac{U}{v_1^2}, \text{ nearly,}$$

when the speed of rim, v, is expressed in feet per second, and U in foot-pounds.

German engineers use the following coefficients, $\Delta E \div \int pds$ being taken for a single stroke, instead of a revolution of the engine :

Engine without expansion.

$n = \frac{l}{r}$	Single crank.	Two cranks.	Three cranks.
4	0.2717	0.1672	0.0693
5	0.2577	0.1422	0.0594
6	0.2489	0.1256	0.0504
7	0.2489	0.1136	0.0453
8	0.2384	0.1046	0.0414
Infinite	0.2105	0.0422	0.0181.

Engine with expansion. Steam cut off at—

	$\frac{2}{3}L$	$\frac{1}{2}L$	$\frac{1}{3}L$	$\frac{1}{4}L$	$\frac{1}{6}L$	$\frac{1}{8}L$
Single crank.	0.3741	0.4076	0.4372	0.4523	0.4625	0.4702
Two cranks	0.2044	0.2250	0.2412	0.2496	0.2552	0.2594
Single crank	0.3252	0.3594	0.3916	0.4088	0.4216	0.4300
Two cranks.	0.0652	0.0720	0.0786	0.0820	0.0846	0.0862

Where, as often in rolling-mill work, the function of the wheel is the storage and restoration of energy to meet fluctuations of the work of resistance, mainly, the values of ΔE and of U become the measure of that work as a maximum per revolution of the engine, or for a succession of revolutions comprising a "cycle" which continually or frequently repeats itself, reproducing in order the variable conditions. In such a case the value of m is taken as that of the fluctuation allowable during such cycle, and the duty of the engine consists largely in storing power during periods of minimum for use at maximum draught of energy.

It is for this reason that rolling-mill engines, when their wheels are the only regulators, are fitted with fly-wheels of several times as much energy as would ordinarily be supplied. Such wheels are sometimes placed on a separate "jack-shaft," and geared up to as high a speed of rotation as is safe.

In this, as in all other cases, the energy to be stored and

restored must be determined and the range of speed of rotation allowable settled; *the stored* energy is then always made

$$U = \frac{W(v_2^2 - v_1^2)}{2g}; \quad W = \frac{2gU}{v_2^2 - v_1^2}.$$

The strength of the fly-wheel being taken as limited by that of its rim, in resistance of centrifugal force, the maximum velocity safely attainable is determined by either the strength, in tension of its arms, by the strength, in the tangential line, of the weakest point in the rim, which is commonly the connection, in large built-up wheels, of section with section; or both combined. For small wheels, the second only is generally cal culated upon; for large wheels the first only. It is not usually safe to depend on their effective combination.

The centrifugal force tending to fracture the rim of the fly-wheel is computed thus:

Let R = the radius at the centre of gyration of its section;
s = area of section;
α = angle made by any radius considered, with the plane of separation;
w = the weight of unity of volume;
ω = the angular velocity of rotation.

Then the weight of an elementary arc is $\overset{W}{\cancel{w}}sRd\alpha$ and its centrifugal effort is

$$\frac{wsRd\alpha}{g}\omega^2 R = \frac{wsR^2\omega^2}{g}d\alpha = Fd\alpha.$$

Its direct and lateral components are

$$F_a = F\int_0^{\pi} \sin\alpha d\alpha = 2F = \frac{2Ws\omega^2 R^2}{g}; \quad F\int_0^{\pi} \cos\alpha d\alpha = F_b = 0.$$

This quantity measures the force which tends to divide the wheel-rim on a diametral plane, and which is sustained by the two sections of the solid rim or by their connecting pieces, bolts or other. Each section or the fastenings on one side must resist one half this stress.

In the case of the "parallel-rod" or "side-rod" of the locomotive, this computed total stress is shared by the two pins by which it connects, in addition to or reduced by the weight of the rod, accordingly as it is in the lower or the upper portion of its path. It will be found by computation, in the latter case, that this stress is, at high speeds, fifteen or even twenty or more times the weight, and may amount to two or three tons. In such a case careful counterbalancing is resorted to.

To determine the stress on the arms, let W be the weight of each section carried by an arm of radial length, r. Then the radial pull is, nearly,

$$F = \frac{Wv_1^2}{gr} = \frac{2Wv_1^2}{gD}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the section must be such that, dividing F by its area, the stress sustained by unity of such area is a safe one—not exceeding, we may say, one-eighth the strength of the metal.

If the rim is continuous and its strength is relied upon to give safety against fracture at high speeds, the radial effort of unity of volume, of the density, w', is, for a wheel of diameter D,

$$F' = \frac{2w'v_1^2}{gD}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and this, at the instant of breaking, is resisted by a tensile force of equal amount, or

$$F' = \frac{2T}{D} = \frac{2w'v_1^2}{gD},$$

and

$$v_1 = \sqrt{\frac{gT}{w'}}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

T being the tenacity of the material. Thus the allowable velocity of the rim varies as the square root of the quotient: tenacity of material divided by the density of the metal.

Since the linear speed of a solid rim, as a maximum, must not exceed that giving maximum safe stress on the iron of which it is composed, it is thus obvious that this maximum velocity will be

$$v = a \sqrt{t},$$

where the safe stress, t, of the metal is given. The value of a, in British measures, is nearly 3; it is 0.035, nearly, in metric measures; the units giving v in feet per second and t in pounds per square inch, in the one case, and in metres per second and kilograms per square metre in the other. A fly-wheel in segments must be computed with reference to the stresses coming on the parts intended to sustain them.

For a maximum safe stress of 10,000 pounds per square inch, or 1,440,000 per square foot, as for wrought-iron or steel, the breaking speed is found at about 300 feet per second, or 3.5 miles per minute, nearly. Taking one half that figure for cast-iron, the maximum speed becomes about 220 feet per second, or nearly 2½ miles per minute; and about one third these velocities are usually taken as safe. The safe speeds are, respectively, not far from 200 and 150 miles per hour.

The tension at any section of the rim is, the units being pounds, feet, and minutes,

$$T = 0.00034 \frac{WN^2R}{2\pi}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and this is usually made less than 3000 pounds per square inch for cast-iron.

The volume, V, of the rim of the wheel must be equal to the demanded weight, W, divided by the density of the metal; and its cross-section will be that volume divided by its mean circumference. Thus, taking the weight of cast-iron at 450 pounds per cubic foot, its section is, in square feet,

$$A = \frac{V}{\pi D} = \frac{W}{450\pi D} = 0.0005\,\frac{W}{D}, \text{ nearly.}$$

The form of section is decided by exterior conditions; but it is usually rectangular in large wheels, often circular or elliptical in small constructions.

The strength of the arms to resist bending stress is determined by calculating them as "cantilevers," or beams secured at one end and loaded at the other.

112. **Determinations of the Form of Wheel,** and its proportions of diameter and weight, are made after considering the conditions of operation and of location. As a rule, the diameter may be made either great or small, and the weight of wheel-rim correspondingly diminished or increased, as the designer may choose; but a very large wheel occupies valuable space, may interfere with the construction of the engine-foundation, and compels a high linear velocity of rim, which may, in some cases, approximate the limit of resistance to centrifugal forces.

A diameter of about 4 times the engine-stroke is often found satisfactory; but the higher the speed of engine, the less the diameter permissible. Engines of great length of stroke and of low speed of revolution, in the early half of the nineteenth century, had wheels thirty or forty feet in diameter; but with the higher velocities now common, the same size of cylinder is employed to produce several times the power; the speeds of rotation have been increased correspondingly, and the diameter of wheel rarely exceeds 18 or 20 feet with the

largest engines. The proportions of the wheel, at the same time, have been greatly changed, the rim being heavier in proportion to diameter, and much better regulation is sought.

To find the mean radius of the fly-wheel rim:

Let r = any radius;
r_1 = external radius;
r_2 = internal "
r_m = mean "

Then the moment of inertia is

$$M = 2\pi\int_{r_2}^{r_1} r^2 r dr = 2\pi\int_{r_2}^{r_1} r^3 dr;$$

$$= \tfrac{1}{2}\pi(r_1^4 - r_2^4) = Sr_m^2; \quad S = \pi(r_1^2 - r_2^2);$$

$$r_m = \sqrt{\tfrac{1}{2}(r_1^2 + r_2^2)}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and if the diameter, D, be measured as $2r_m$,

$$D = 2r_m = \sqrt{2(r_1^2 + r_2^2)}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

113. **Fly-wheels for External Work,** as has been already shown, must be specially proportioned to meet the case in hand, and cannot be, in all cases, given the proportions suitable for the engines driving them. In a heavy rolling-mill, for example, a wheel weighing a hundred thousand pounds may be slowed down from its full speed of fifty revolutions per minute, to half that velocity in a fraction of a minute; while, as a regulator of the engine, the extreme fluctuations would be quite insensible.

In all such cases the work to be done, and energy stored and restored, must be computed; the time allowable and range of speed permissible in such transfer of energy must be settled upon; and the maximum diameter and speed of wheel-rim fixed. The weight of metal needed in the rim to store

and restore that quantity of work and energy, under the prescribed conditions, may then be easily computed. In such cases the fluctuation of speed permitted is generally very great, often as high as $\frac{1}{m} = 0.2$, or more, and the weights of wheel are thus rendered comparatively small. It should also be remembered, when making such computations, that the engine is usually so constructed that it will enormously increase its effort, when the wheel is thus slowing down, through the operation of its governor and decreased expansion; but this action should not be relied upon, in heavy rolling-mill work, to reduce the weight of the wheel very greatly.

114. **Inertia and its Effects** enter into the theory of the high-speed engine to a much more important extent than with the engine of slower movement. The great velocity of rotation gives the balance-wheel a correspondingly enhanced effectiveness, and insures a greater uniformity of motion, both from instant to instant and minute by minute. The wheel turns the more steadily for the accelerated velocity, and the governor, with its increased power and quickness of action, preserves so much the more exactly the mean speed of the engine. The action of reciprocating parts, in modifying the pressures on the running parts, and on the crank-pin and other journals, is intensified, and, from being insignificant, it is made to take position as one of the most important and serious of all the forces acting in the machine. The energy stored at any instant is measured by the product of the mass into the half-square of the velocity, and the intensity of the force due its storage or its restoration is, as a mean, the quotient of the variation of this quantity of energy by the distance through which that force acts to produce its effect.

Thus the regulating action of the wheel or the pressure-distributing power of the reciprocating parts becomes a rapidly varying function of the speed of engine, intensifying the effect of the balance-wheel, and modifying crank-pin pressures more and more sensibly as the speed of engine increases. At the velocities of rotation now common, the size and weight of

wheel are reduced enormously as compared with the dimensions found desirable at once maximum speeds; and the pressures on the pin are very greatly affected, becoming comparatively uniform throughout the stroke.

The speeds of piston of small, fast-running engines often now approximate, and sometimes greatly exceed, $500 \sqrt[3]{s}$. Thus a small engine by well-known builders * is worked up to the following speeds:

3-inch	stroke,	870	revolutions,	435	feet	per	minute.
4	"	760	"	510	"	"	"
5	"	680	"	570	"	"	"
6	"	620	"	620	"	"	"
9	"	500	"	750	"	"	"
12	"	440	"	880	"	"	"

The large marine engines employed in vessels of great speed, such as are now common in the transatlantic service, are continually coming more and more generally to this standard; while locomotive practice sometimes illustrates even higher figures. At such speeds of rotation and of reciprocating parts as are thus adopted, the effects of inertia become not only sensible, but often of very serious magnitude. In the case of the City of Paris, for example, the weight of reciprocating parts was, in the low-pressure engine, about 35,000 pounds; while the stress due their inertia, when "turning the centre" at a speed of 90 revolutions per minute, was not far from 125 tons, and was quite sufficient to destroy the engine, in case of a broken shaft, at something less than four times this maximum working speed.

The result of the action of such forces, in the regular operation of the engine, are traceable both outside the engine and within its structure. The external effects are seen in a tendency to shake the engine on its foundations, to produce variation of stresses in every line intersecting the centre of the crank-shaft, and to throw out of balance parts which would

* Proc. Inst. C. E., 1885–6, p. 170.

otherwise work with steadiness and revolve smoothly. It is this which dictates careful balancing, where practicable, of all running parts, whether reciprocating or rotating.

The effects of inertia-forces, within the engine, are thus seen in a modification of the pressures on the several parts and their connecting-pins, and especially on the crank-pin, where it often happens that the pressures are actually widely different from those which are due the action of the steam-pressure alone. The general tendency is evidently to decrease the pressures at the moment of beginning the stroke and to correspondingly increase them toward the end; the steam-pressure being often largely, and sometimes wholly, absorbed in overcoming the resistances due acceleration while an equal and opposite force is exerted at the end of the stroke by the same parts, then subjected to forcible retardation, and at a time when the steam-pressure may have become reduced by expansion to a very small amount. These effects may be utilized in the adjustment of pressures on the crank-pin, in such a manner as to produce decidedly advantageous resultant effects. This was first effectively and philosophically attempted by Mr. Porter, and with great success.*

The computation of the inertia-effects are easily made by algebraic and graphical methods; and Mr. E. F. Williams has devised a simple instrument to produce automatically records of their magnitude and variations in engines in actual operation.†

Professor Jacobus has computed by the exact, and by the usual approximate and simple, methods elsewhere referred to, the following quantities as illustrating these representative cases in practice (Table, p. 428).‡ It is seen that the latter are amply exact for all ordinary purposes of the engineer.

115. **Rotating Efforts** and turning moments, as felt at the crank-pin, inertia being neglected or insensible, vary greatly in different parts of the orbit of the pin and for correspondingly

* C. T. Porter on the Richards Indicator.

† Am. Machinist; 1884.

‡ Trans. Am. Soc. M. E.; vol. XI; 1889; No. CCCLXXVII.

BALANCING STANDARD ENGINES.

Class of Engine.	Conditions assumed.	Dimensions. Bore.	Dimensions. Stroke.	Revolutions per minute.	Indicated horse-power.	$\frac{\Delta E}{\int P ds}$ Exact.	$\frac{\Delta E}{\int P ds}$ Approximate.	$\frac{\Delta E}{\int P ds}$ Not including accelerating forces.	Pressure acting on crank-pin. Lbs. per sq. in. Exact.	Pressure acting on crank-pin. Lbs. per sq. in. Approximate.	Pressure acting on crank-pin. Lbs. per sq. in. Not including accelerating forces.
		Inches	Inches								
Small horizontal high-speed..............	Not including the effects of friction and weight..	10	12	300	57	.184	.174	.238	58.1	57.3	83.0
Small horizontal high-speed..............	Not including friction, but including weight.......	10	12	300	57	.180	....	....	58.0	....	....
Small horizontal high-speed..............	Including both friction and weight................	10	12	300	57	.171	....	....	56.8	....	....
Locomotive..............	Not including friction and weight................	18½	.24	250	*345	.181	.172	.275	56.8	55.5	108.0
Corliss. Slow speed of revolution...........	Not including friction and weight................	26⅛	60	60	346	.142	.131	.185	76.1	75.5	89.4
Westinghouse...........	Not including friction and weight................	11	10	320	66	.160	.147	.247	61.0	60.8	75.5

* For one cylinder.

various positions of the piston. At the "dead-centre" they are zero, whatever the pressure on the piston; they become maxima at points in the crank-pin circle which are determined by a combination of the kinematic relations of crank-path to its connecting-rod and of the latter to the piston movement and of the variation of steam-distribution before and behind the piston. These several influences have been carefully studied by many writers, and have not only been completely analyzed, but the still other complications introduced by the action of gravity and of friction have been brought into the problem and a solution effected.

For a complete and exact demonstration reference must be made to the various original papers which contain accounts of such mathematical researches.* Their value to the engineer depends upon their efficiency in aiding him to proportion the parts of his engines in such manner as to insure smooth running and durability at maximum speeds.

To determine the equivalent tangential effort on the crank-pin, that on the piston being given:

Let P be the pressure on the piston;
P_1 the equivalent tangential pressure at the pin;
r the radius of the crank;
l the length of the connecting-rod;
θ the angle which the crank makes with the horizontal.

Then, while the pin moves through $r\theta$, the piston moves:

$$r - r\cos\theta \mp (l - \sqrt{l^2 - r^2\sin^2\theta})$$

for the $\left\{\begin{matrix}\text{forward}\\\text{backward}\end{matrix}\right\}$ stroke.

---

* Disturbing Forces in Locomotive Engines; A. W. Makinson, Proc. Brit. Inst. C. E., 1862–3. Papers by Prof. F. Jenkins, Trans. R. Soc. Edin.; 1879; vol. XXVIII. Reciprocation in High-speed Engines; Arthur Rigg; Iron, Apr. 23, 1886, p. 362. Distribution of Unequal Pressures; J. C. Hoadley; Trans. A. S. M. E., 1880. Formulæ and Tables; Geo. I. Alden; *Ibid.* 1886. Notes in Engineering; G. Lanza; Boston, 1886. Dynamics of Reciprocating Engines; M. G. Cooley; Technic; 1888. General Solution; D. S. Jacobus; Trans. A. S. M. E.; Nov., 1889.

Let θ be increased by $\delta\theta$; then while the pin moves through $r\delta\theta$, the piston will move through

$$d.\left[r - r\cos\theta \mp (l - \sqrt{l^2 - r^2\sin^2\theta})\right]$$
$$= \left(r\sin\theta \mp \frac{r^2\sin\theta\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right)\delta\theta,$$

and

$$P\left(r\sin\theta \mp \frac{r^2\sin\theta\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right)\delta\theta = P_1 r\delta\theta;$$

then

$$P_1 = P\left(\sin\theta - \frac{r\sin\theta\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right)$$

for the forward stroke, and

$$P_1 = P\left(\sin\theta + \frac{r\sin\theta\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right)$$

for the backward stroke.

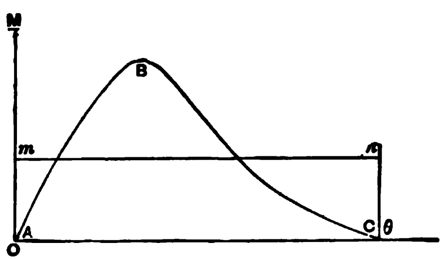


FIG. 168.—TWISTING MOMENTS.

The product of the tangential effort on the pin into the crank-radius give the measure, at each instant, of the turning-moment. Were the motion precisely harmonic, these values being plotted, for uniform pressure, the curve of twisting-mo-

ments, or of "torque," would be symmetrical and a curve of sines, of which the equation (l infinite) would evidently be

$$M = Pr \sin \theta = P_1 r.$$

The true curve, however (l finite), is slightly unsymmetrical and in usual cases, in which steam-pressure is constantly varying, substantially as in the illustration, in which the expansion is taken into account. This diagram (Fig. 168) is made by obtaining each ordinate from the indicator-diagram, measuring the effective pressure at the appropriate crank-position, and inserting its value in the equation for the torque. The mean value of this quantity is Om, the rectangle measuring an area equal to that under the curve as obtained. In engines having two or more cranks, these curves being superposed properly, the resultant irregularities, becoming less and less with their larger number, may be easily measured, and will be found comparatively unimportant.

116. **The Inertia of Reciprocating Parts,** assuming truly harmonic motion of crank and piston, is easily and simply determined thus:

Let ϕ = angle of crank with path of piston;
s = distance of piston from end of stroke;
R = length of crank;
v = velocity of piston;
t = time;
T = time of one revolution;
a = angular velocity;
f = acceleration-~~rate~~ in terms of velocity;
F = acceleration-force in terms of weight
$F = f\dfrac{W}{g}$.

Then

$$a = \frac{d\phi}{dt} = \frac{2\pi}{T} = \text{constant};$$

$$s = R(1 - \cos at) = R(1 - \cos \phi);$$

$$ds = Rd \cdot \text{versin}\ \phi = aR \sin(at) . dt = R \sin \phi d\phi; \quad . \quad (1)$$

$$v = \frac{ds}{dt} = \frac{R \sin \phi d\phi}{dt}$$

$$\therefore \ f = \frac{dv}{dt} = \frac{R \cos \phi d\phi^2}{dt^2};$$

$$= R \cos \phi \frac{4\pi^2}{T^2};$$

$$= a^2 R \cos \phi, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

since $d\phi$ and dt are constant. Make

$$\cos \phi = \pm 1;$$

then

$$f_m = \pm a^2 R = \text{max.} = \text{centrifugal force} = \frac{4\pi^2}{T^2} R. \quad . \quad (3)$$

But

$$F_{max} = f_m \frac{W}{g} = \frac{4\pi^2}{T^2} \cdot \frac{RW}{g} = \frac{\pi^2 RW}{8.0416 T^2},$$

as due to centrifugal force.

Generally, we have

$$F = Mf = \frac{W}{g} f = \frac{W}{g} a^2 R \cos \phi = \frac{WR\pi^2}{8.0416\, T^2} \cos \phi;$$

or

$$= 1.223 \frac{RW}{T^2} \cos \phi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

To determine the intensity, p, of the equivalent pressure on the piston, we have

$$pA = F = 1.223 \frac{RW}{T^2} \cos \phi = \frac{1.223 N^2 RW \cos \phi}{3600};$$

where N = revolutions per minute,

$$p = 0.000341 \frac{N^2 RW}{A} \cos \phi; \quad . \quad . \quad . \quad . \quad . \quad (5)$$

when $\phi = 0$, and the crank is on the centre,

$$p_{max} = 0.000341 \frac{N^2RW}{A}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which expressions R = radius of crank in feet, W = weight of reciprocating parts in pounds, A = area in square inches, and p = pounds per square inch.

For radius in metres, weight in kilograms, pressure in kilograms on the square centimetre, areas in square centimetres,

$$p^m_{max} = 0.0072 \frac{N^2R_mW_m}{A_m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Since $F \propto R \cos \phi$, i.e., $F \propto R \cos at$, and since the distance of the piston from the half-stroke position is, neglecting the angularity of the rod, equal to $R \cos \phi$, the intensity of the accelerating or retarding effort is proportional to the latter distance; and we have the law illustrated thus, as it is evident that the method of variation is the same both in the storing and the restoring of work.

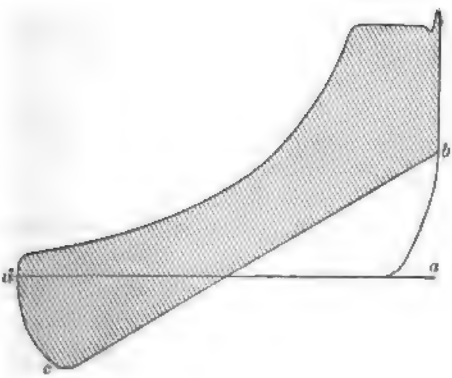


FIG. 169.—ACCELERATING FORCES.

The resistance to acceleration, or effort due retardation, in the steam-engine being taken as measured by the expression

$$P \doteq 0.000341\, N^2pR;$$
$$= 1.226n^2pR; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where N and n are the number of revolutions per minute and per second; p is the pressure per square inch of piston due the total weight of running parts producing that effort or resistance; and R is the crank-radius. The following table gives the pressure per square inch of piston due to the acceleration or retardation when the total weight of running parts is one pound per square inch of piston-area:

Crank R =	1	2	3	4	5
Revolutions per minute, N. 25	0.22	0.43	0.64	0.85	1.06
50	0.86	1.70	2.54	3.40	4.25
75	1.92	3.84	5.75	7.65	9.56
100	3.40	6.80	10.15	13.10	16.38
150	7.65	15.35	22.95	30.53	38.19
200	13.60	27.20	40.50	54.40	68.10
250	21.25	42.50	63.75	45.00	106.25
300	30.60	61.20	91.80	122.40	153.00

These figures being multiplied by the quotient, weight of running parts divided by piston-area, will give approximately the forces acting with or against the steam-pressure.

The effect of this action in modifying the pressures on the crank-pin is seen in the last figure, in which

$$ab = p = 0.000341\frac{N^2}{R} = cd\cdot$$

represents the quantity by which those pressures are reduced at the beginning and increased at the end of the stroke, in such manner as to render them much more uniform throughout the stroke. At mid-stroke, with a long connecting-rod, the acceleration effect becomes zero and reverses its sign, becoming a reinforcement of the steam-pressure instead of, as previously, an opposing force. By a careful adjustment of weights to speeds a very useful regulating action may be thus introduced, and one which, in its own way, is superior to that of the fly-wheel, which does little to save the crank and shaft from shocks.

When the motion is not harmonic, as when the connecting rod is considered in its effect on the motion of the piston and its rod, we obtain the following closer approximation :

Let ϕ = angle traversed by the crank;
θ = angle of the rod with line of centres;
r = crank radius;
l = length of rod;
$m = l \div r$.

Then, since θ is always small, we may take $\tan\theta = \tan\phi$, and then

$$\sin\theta : \sin\phi :: r : l;$$

$$\sin\theta = \frac{r}{l}\sin\phi = m\sin\phi.$$

But the ratio of velocities of crank-pin and piston, $\frac{v'}{v} = n$, is

$$n = \sin\phi + \cos\phi\tan\theta;$$

$$= \sin\phi + \frac{1}{n}\cos\phi\sin\phi;$$

$$= \sin\phi + \frac{1}{2n}\sin 2\phi;$$

whence the acceleration becomes

$$f = \frac{dv}{dt} = v'(\cos\phi + n\cos 2\phi)\frac{d\phi}{dt}.$$

Since $v' = r\dfrac{d\phi}{dt}$,

$$f = \frac{dv}{dt} = \frac{v'^2}{r}(\cos\phi + n\cos 2\phi),$$

and the effort due this acceleration is

$$F = W\frac{f}{g} = \frac{Wv'^2}{gr}(\cos\phi + n\cos 2\phi),$$

when W is the weight of the parts affected by the acceleration; and this quantity divided by A, the area of piston, gives the equivalent intensity of pressure to be deducted from or added to that of the steam as shown on the indicator-diagram, as

$$p' = \frac{F}{A},$$

$$= \frac{Wv'^2}{Agr}(\cos\phi + n\cos 2\phi).$$

At each end of the stroke, $\phi = 0$, and $\phi = 180°$, $n = 0$;

$$p' = \frac{Wv'^2}{Agr}; \quad \text{or} \quad p' = \frac{Wv'^2}{Agr}$$

At "half-centre," $\phi = 90°$, $\cos 2\phi = 1$, and $\cos\phi = 0$;

$$p' = n\frac{Wv'^2}{Agr}.$$

When $p' = 0$, the line on the diagram cuts the axis of X, and

$$\cos\phi = \frac{1}{4}\left(\sqrt{\frac{1}{n^2} + 8} - \frac{1}{n}\right).$$

In these expressions it is assumed that *all* the reciprocating masses may be taken as at the piston and included in W.

The tangential effort at the pin has the ratio to the effort at the piston

$$\frac{p_t}{p} = n' = \frac{\sin(\phi + \theta)}{\cos\theta},$$

and

$$p_t = n' p_e = p_e \quad \frac{\sin(\phi + \theta)}{\cos\theta}.$$

But the pressure reaching the pin is the steam-pressure, less p', and thus

$$p_t = (p - F)\frac{\sin\phi + \theta}{\cos\theta}$$

$$= p - \frac{Wv'^2}{gr}(\cos\phi + n\cos 2\phi)\frac{\sin(\phi + \phi)}{\cos\theta};$$

and taking

$$\frac{\sin(\phi + \theta)}{\cos\theta} = \sin\phi + \tfrac{1}{2}n\sin 2\phi,$$

$$p_t = \left[p - \frac{Wv'^2}{gr}(\cos\phi + n\cos 2\phi)\right](\sin\phi + \tfrac{1}{2}n\sin 2\phi).$$

Taking, as a common value, two pounds per square inch of piston for the weight of the reciprocating parts, the following are the corresponding pressures per square inch of piston due to their inertia forces : *

	Crank-radius : feet	1	2	3	4
Rev. per min.	50	1.71	3.40	5.08	6.8
	75	3.83	7.67	11.5	15.3
	100	6.80	1.36	2.03	27.2
	150	15.3	30.7	45.9	61.1
	200	27.2	34.4	81.1	
	250	42.5	85.0		
	300	61.2			

* See Kennedy's Mechanics, p. 351.

Thus inertia-forces, insignificant at low speeds, become important at those now becoming usual. In an engine of familiar type, at 1000 revolutions per minute, this pressure may be over 110 pounds, at the weight above assumed.

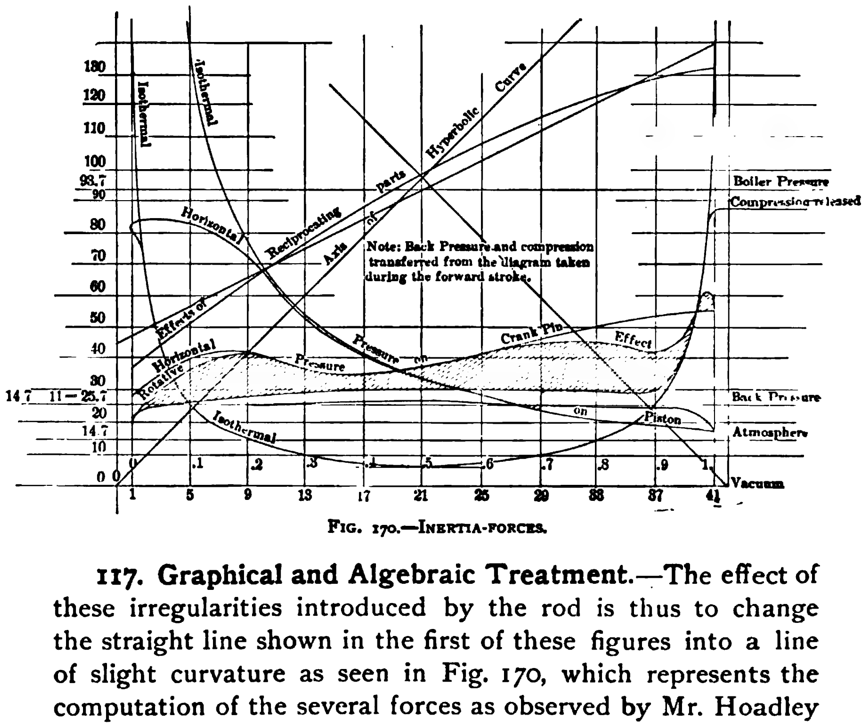


FIG. 170.—INERTIA-FORCES.

117. Graphical and Algebraic Treatment.—The effect of these irregularities introduced by the rod is thus to change the straight line shown in the first of these figures into a line of slight curvature as seen in Fig. 170, which represents the computation of the several forces as observed by Mr. Hoadley in the operation of an engine of 24 inches diameter of cylinder, 3 feet stroke, making 179 revolutions per minute, the weight of reciprocating parts being 1663 pounds.* The shaded areas exhibit the *net* rotative effect, and show that more work is done in the last than in the first quadrant. The load is light and the back-pressure unusually great. The diagram is fully self-explanatory.

A simple method of approximately locating the curve which takes the place of the straight line, determined on the

* Letter to Am. Machinist; June 7th, 1884.

assumption that the angularity of the connecting-rod may be neglected, is the following:

Three points are found, and a circular arc drawn through them will closely coincide with the true parabolic line. The straight line for the first case being ABC, the ordinates OA and XC are known, and B is the middle of OX. In the new line, E is that point at which the piston stands when the line of the connecting-rod is tangent with the crank-pin orbit. At

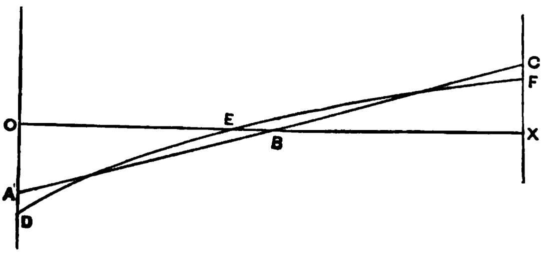


FIG. 171.—INERTIA-CURVE.

either end the actual magnitude of centrifugal force is less than that computed for weights at the crank-pin in the proportion of the radii measured, in each case, from the instantaneous axis, i.e., in the proportion of crank to rod. Hence if the rod be five times the length of the crank, a usual and good proportion, reduce the ordinates OD and XC each by one fifth, and the new ordinates, OA and XF, are those of the extremities of the curve DEF, which was to be found.

An exceedingly simple and convenient graphical construction determining the acceleration of reciprocating parts is due to Dr. Rittershaus.*

In Fig. 172 let the crank AB, the rod BC, and the engine line AC be represented as there shown; let AZ be ~~drawn perpendicular to~~ [illegible] CZ to AC, and AD to AC, DE to CD, and make DF and BE parallel.

Then Z is the instantaneous axis of BC, and the velocities of B and of C have the ratio

$$\frac{v}{V} = \frac{CZ}{BZ}$$

* Civil ingenieur, xxv; also Unwin, Machine Design, § 280.

Produce CB to meet AD. Let r = the crank-radius, $M = AD$. Then

$$\frac{v}{V} = \frac{M}{r}; \quad v = M\frac{V}{r};$$

$$M = AC \tan \alpha = x \tan \alpha;$$

$$\frac{dM}{dt} = \frac{dv}{dt} \cdot \frac{r}{V} = x \frac{1}{\cos^2 \alpha} \cdot \frac{d\alpha}{dt} + \frac{dx}{dt} \tan \alpha.$$

Since $$\frac{dx}{dt} = v = M\frac{V}{r},$$

$$f = \frac{dv}{dt} = \frac{V}{r} \cdot \frac{x}{\cos^2 \alpha} \cdot \frac{d\alpha}{dt} + M\frac{V^2}{r^2} \tan \alpha.$$

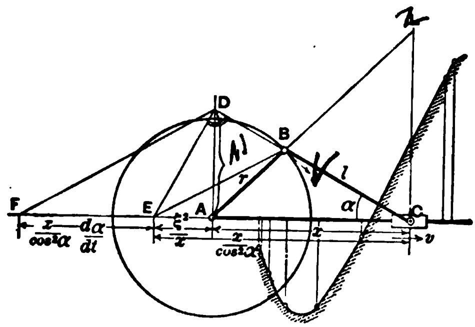


FIG. 172.—ACCELERATION-DIAGRAM.

If we introduce a velocity $-v$ to bring C to rest, the angular velocity of the connecting-rod about C is

$$\frac{d\alpha}{dt} = -\frac{v}{M}\frac{DB}{BC} = -\frac{V}{r}\frac{DB}{BC},$$

and

$$f = \frac{dv}{dt} = -\left(\frac{V}{r}\right)^2 \left\{ \frac{x}{\cos^2 \alpha}\frac{DB}{BC} - M \tan \alpha \right\};$$

and if we make $V = r$,

$$\frac{dv}{dt} = -\left\{\frac{x}{\cos^2 \alpha}\frac{BD}{BC} - M\frac{M}{x}\right\}.$$

But we have made

$$EC = \frac{x}{\cos^2 a}; \quad \frac{FE}{EC} = \frac{DB}{BC};$$

$$FE = \frac{x}{\cos^2 a}\frac{BD}{BC}.$$

Again,

$$\frac{EA}{M} = \frac{M}{x},$$

$$EA = \frac{M^2}{x},$$

whence

$$f = -\{FE - EA\}.$$

For the dead-points, we have for the inner one

$$x = l + r \quad \text{and} \quad \frac{DB}{BC} = \frac{r}{l};$$

for the further,

$$x = l - r \quad \text{and} \quad \frac{DB}{BC} = -\frac{r}{l}.$$

Hence, for the inner,

$$f_1 = \frac{dv}{dt} = -(l + r)\frac{r}{l} = -r\left(1 + \frac{r}{l}\right);$$

for the outer,

$$f_2 = \frac{dv}{dt} = (l - r)\frac{r}{l} = r\left(1 - \frac{r}{l}\right)$$

If W be the weight of the parts, A the area of the piston, the accelerating force, per unit of area, is

$$\frac{Wf}{gA}.$$

We thus measure the values of f on a scale for which $v = r$, multiply them by $\frac{W}{gA}$, and obtain values of ordinates of the acceleration-curve.

The following examples illustrate the application of the preceding principles in actual cases: *

The disturbing influence of the angularity of the connecting-rod and of the inertia of the reciprocating parts can be located by kinematics, and measured by graphics, much more readily and speedily than by the processes of trigonometry. It is proposed to show how the proper relation between the pressure on the piston and the tangential pressure on the crank-pin as so modified can be delineated.

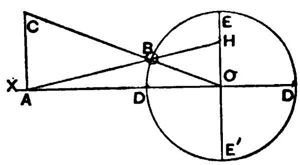


FIG. 173 —MOTION OF ENGINE.

In Fig. 173, let O be the centre of the shaft, OB the crank, and AB the rod.

The cross-head A has a reciprocating motion along OX, and the crank-pin B a continuous uniform motion in the circumference $DBED'E'$.

Draw AC perpendicular to OX, and produce the crank-line OB until it intersects AC. The point of intersection, C, gives

* By P. A. Engr. A. B. Canaga, U. S. N., Assist. Prof. M. E., Sibley College, Cornell University; 1889. See Journal Am. Soc. Naval Engrs., Nov. 1889.

the instantaneous axis of the rod for its position AB. Produce AB to intersect the vertical diameter $E'E$ in H. The intercept OH is to OB as the velocity of the cross-head or piston is to the velocity of the crank-pin; for, if

$V =$ velocity of the cross-head or piston, and
$v =$ velocity of the crank-pin, then

$$AC : BC :: V : v. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In the triangles ABC and OBH, equiangular and similar,

$$AC : BC :: OH : OB; \quad . \quad . \quad . \quad . \quad . \quad (2)$$

combining equations (1) and (2) gives

$$OH : OB :: V : v. \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Let $P =$ intensity of pressure on the piston;
$P' =$ total tangential pressure on the crank-pin divided by the area of the piston.

We shall then have

$$PV = P'v, \text{ or}$$
$$P' : P :: V : v, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and

$$OH : OB :: P' : P. \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

In Fig. 173, if OB be drawn to scale to represent the *pressure* per square inch on the piston, OH will represent to the same scale the tangential pressure on the crank-pin due to the pressure P per square inch of piston.

If, on the other hand, for equation (3), OB be drawn to represent to scale the *velocity* of the crank-pin in its orbit, OH will represent to the same scale the velocity of the piston, and Fig. 173 also a graphical expression of equation (3).

Assume the following data:

Engine, horizontal, direct-acting.
Diameter of piston, 20 in.
Stroke of piston, 3 ft.
Length of connecting-rod $= 4 \times$ crank $= 4 \times 1\frac{1}{2} = 6$ ft.
Revolutions per minute, 160.
Weight of reciprocating parts, 1006 lbs.
Weight of reciprocating parts per sq. in. of piston $= 1006 \div \overline{10}^2\pi = 3.2$ lbs. $= W$.
Velocity of crank-pin per second, $3 \times \pi \times 160 \div 60 = 25.13$ ft.
Scale of indicator, 1 in. $= 40$ lbs.
The velocity of the crank-pin is regarded as constant.

With O as a centre, in Fig. 174, and a radius OD to some convenient scale,—say, $\frac{1}{12}$ of an inch,—to represent one foot of the velocity per second of the crank-pin in its orbit, describe the circle $DED'E'$. Given in this scale $OD = \frac{25.13}{12} = 2.09$ in.

Divide the circumference of the crank-pin circle into an even number of equal parts, as 20,* as this number gives 10 ordinates of the diagrams for each stroke. Draw $D'DX$ to represent the direction of motion of the piston. From D and D' lay off Dd and $D'd'$, respectively equal to four times the crank OD; d and d' will represent the ends of the stroke of the piston. From the points of division on the circumference 1, 2, 3, 4, etc., as centres, and a radius $= 4 \times OD =$ length of connecting-rod to scale, describe arcs cutting OX in the points $1'$, $2'$, $3'$, $4'$, etc. These latter points will give the positions of the pistons when the crank-pin is at the positions 1, 2, 3, 4, etc., respectively. The velocity of the crank-pin being constant, the intervals of time required for the crank-pin to pass these equal

* The $\frac{1}{20}$ of an arc of a complete circle can be readily found by laying off from D on the diameter DD', Fig. 178, the distance $DL = \frac{1}{10}DD'$. With D and D' as centres and the diameter DD' as a radius, describe the arcs intersecting at X. From X draw the line XL, producing it to B; then the arc DB will be approximately the $\frac{1}{20}$ of the circumference. The distance DB can be used in the spacing dividers for stepping off the 20 divisions.

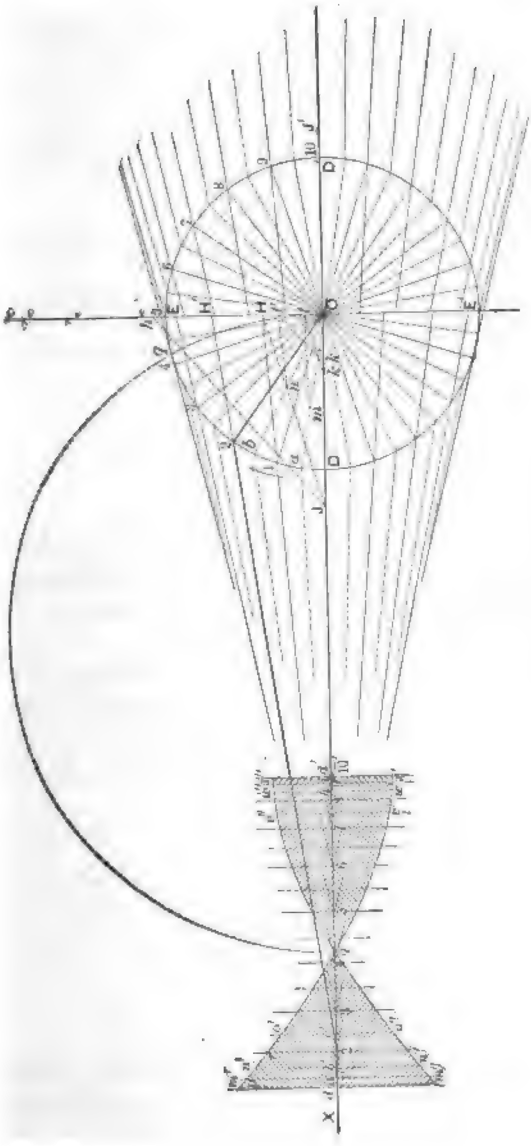


FIG. 174.—ENERGY OF PARTS.

divisions of its orbit will be equal, and these small intervals of time will be to the second of time as the small spaces in the divisions of the crank orbit are to the velocity of the crank-pin per second. When the crank-pin is at D, the reciprocating parts have a horizontal velocity $= 0$, but when it arrives at position 1, the reciprocating parts have a horizontal velocity, according to equation (3), represented to scale in Fig. 174 by $OH = V_1$.

The energy stored in the reciprocating parts, per square inch of piston-area, is $\frac{WV_1^2}{2g}$, and this energy has been imparted by a piston-pressure P_1 acting through the distance $d1'$. Equating these quantities of work gives

$$P_1 \times d1' = \frac{WV_1^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The mean pressure P_1 is that part of the piston-pressure required to impart the velocity V_1 to the reciprocating parts while the crank-pin moves from D to 1, and while the piston itself travels from d to 1'.

Assume a mean tangential pressure P_1^t acting on the crank-pin through the space $D1$ to impart the velocity to the reciprocating parts that is done by P_1 acting through $d1'$; then the work done by these two pressures is the same; that is,

$$P_1 \times d1' = P_1^t \times D1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Equations (6) and (7) combined give

$$P_1^t \times D1 = \frac{WV_1^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

That is, a mean tangential pressure P_1^t acting from D to 1, a distance equal to $\frac{1}{20}$ of a revolution of the crank-pin $= \frac{1}{20} \times 3\pi = .47$ ft., causes the same energy to be stored in the reciprocating parts as a mean piston-pressure P_1 acting from d to 1'.

The numerical values of the quantities in equation (8) are as follows:

$$W = 3.2 \text{ lbs.};$$

$$2g = 2 \times 32 = 64;$$

$$D1 = \tfrac{1}{20} \times 3\pi = .47 \text{ ft.}$$

P_1^t = mean tangential pressure on the crank-pin, between D and 1, necessary to impart the velocity V_1 to the reciprocating parts. Making these substitutions, equation (8) becomes

$$P_1^t \times .47 = \frac{3.2 \times V_1^2}{2 \times 32},$$

or

$$9.4 P_1^t = V_1^2 = \overline{OH}^2. \quad . \quad . \quad . \quad . \quad . \quad (9)$$

As already stated in drawing Fig. 174, OD was taken so that $\frac{1}{12}$ of an inch represented one foot of the velocity of the crank-pin. A solution of equation (9) would give P_1^t in twelfths of an inch, each of which would represent one pound-pressure per square inch. But as the indicator-scale on which 1 in. = 40 lbs. is to be used to measure pressures, the value of P_1^t is $\frac{40}{12}$ = $3\frac{1}{3}$ times too great. Now, if the constant part 9.4 in equation (9) be multiplied by $3\frac{1}{3}$, and the equality still holding, P_1^t would be reduced to the $\frac{1}{3\frac{1}{3}}$ part of itself, so that the ordinates to be subsequently found by graphics may be measured with the indicator-scale directly. This does not alter the equality of the members of equation (9). Equation (9) now becomes

$$9.4 \times 3\tfrac{1}{3} \times P_1^t = V_1^2 = \overline{OH}^2,$$

or

$$31.33 P_1^t = V_1^2 = \overline{OH}^2. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

In the change from equation 9 to 10 the left-hand member of equation 9 has been multiplied by $3\frac{1}{3}$, and the same member

will be divided by $3\frac{1}{8}$ by using the scale on which 1 in. = 40 lbs. for measuring the value of P_1^t in the following graphical process, so that (10) is in reality a true equation.

Lay off OJ, in Fig. 174, equal to $\frac{31.33}{12} = 2.61$ in.; J will be the fixed point for the graphic solution of all subsequent equations for determining the influence of the inertia of the reciprocating parts. Take $OK = OH = V_1$, and through k draw kt parallel to JH: then $Ot = P_1^t$; for by construction,

$$OJ : OH :: Ok : Ot, \quad \text{or} \quad Ot = \frac{OH \times Ok}{OJ};$$

$$OH = Ok = V_1, \quad \text{and} \quad OJ = 31.33;$$

$$\therefore Ot = \frac{V_1^2}{31.33} = P_1^t. \qquad \ldots \ldots \quad (11)$$

When the crank-pin is at D the tangential pressure is zero, and increases in nearly a constant ratio until the crank-pin arrives at the position 1; when the tangential pressure is such that the mean tangential pressure P_1^t will occur when the crank-pin is near the point midway between the points D and 1; and this approaches more nearly the truth as the number of divisions of the crank-pin orbit are more numerous. Twenty divisions of this orbit give a near approximation for values of P_1^t, P_2^t, P_3^t, etc. In order to find the true equivalent pressure on the piston of the pressure P_1^t, bisect the arc $D1$, Fig. 174, in a. From a with a radius equal to the length of the connecting-rod to scale, find the corresponding position a' of the piston, and consequently the direction $a'a$ of the connecting-rod. Through t parallel to $a'a$ draw a line cutting Oa in m. The distance Om represents to the indicator-scale the mean pressure per square inch on the piston required to accelerate the reciprocating parts while the crank-pin travels from D to 1. From a' lay off $a'm' = Om$, above OX, the line of zero-pressure. This pressure per square inch must be subtracted from the pressure per square

inch on the piston, because its effort is wholly absorbed in imparting motion to the reciprocating parts, and has no immediate tendency to produce rotation of the crank. The equivalent of this pressure was, however, assumed to be acting tangentially on the crank-pin, because of the convenience of kinematic treatment. The inertia-ordinates, $a'm' = Om$, must be added to the piston-pressure ordinates to get the effort transmitted to the pin.

When the pin arrives at position 2, the piston has acquired a velocity $V_2 = OH'$ to a scale of $\frac{1}{10}$ inch to the foot.

The energy in foot-pounds now stored in the reciprocating parts is $\frac{WV_2^2}{2g}$; so that while the crank-pin travelled from position 1 to 2, these parts stored the total stored at position 2 minus that stored at position 1, or

$$\frac{WV_2^2}{2g} - \frac{WV_1^2}{2g}.$$

The energy stored in the reciprocating parts while the pin travelled from position 1 to 2, reduced to an equivalent energy at the crank-pin, is $P_2^t \times .47$. The numerical coefficient remains constant, while P^t varies for each division of the orbit. Putting these quantities of energy into an equation, and multiplying the left-hand member—that is, the one containing P_2^t—by $3\frac{1}{3}$,

$$9.4 \times 3\tfrac{1}{3} \times P_2^t = (V_2^2 - V_1^2) = (V_2 + V_1)(V_2 - V_1),$$

or

$$31.33 P_2^t = (V_2 + V_1)(V_2 - V_1)$$

$$= (OH' + OH)(OH' - OH). \quad . \quad . \quad (12)$$

This equation is solved graphically by laying off $Oh = OH' + OH$, and $Ok' = OH' - OH =$ the distance HH'; then

through k' draw a line parallel to Jh, cutting EE' in t'. The intercept Ot' on EE' will give to the indicator-scale the mean tangential pressure P_2' on the pin between the positions 1 and 2 necessary to impart the increased energy which was stored in the reciprocating parts. Bisect the arc 1–2 in b, and through t' draw a line parallel to the connecting-rod $b'b$ cutting Ob in n; then On will give the length for the inertia-ordinate $b'n'$, which gives to scale the pressure per square inch, on the piston, required to impart the energy of motion to the reciprocating parts while the pin moves from position 1 to 2, and the piston from 1′ to 2′.

The foregoing steps indicate the process to be followed for each of the divisions of the crank-pin orbit, from whence will result a series of ordinates of the inertia-curves $m'n'o'q' - v'w'x'$ and $m''n''o''q'' - v''w''x''$, the former for the forward and the latter for the return stroke. The ordinates of these curves lying above OX must be subtracted from, and those below added to, the corresponding ordinates of the indicator-diagram, giving new ordinates, which will represent the pressures effective in producing a turning effort on the crank-pin. The amount of this pressure which finally becomes tangential pressure at the pin will be determined later.

In the second and third quadrants of the crank-pin circle the connecting-rod itself, instead of its prolongation, cuts $E'E$; this, for corresponding divisions to the right and left of $E'E$, gives shorter intercepts to the right in Fig. 174, and consequently the ordinates of the inertia-curves are shorter at the inboard end of the diagrams. This does not affect the equality of the energy stored and restored during each stroke of the piston.

The energy stored in the reciprocating parts as the pin travels through successive equal divisions decreases from D until the velocity-ratio of piston to crank-pin becomes a maximum, which for the ratio of crank to connecting-rod of

$$\left\{\begin{matrix} 1:6 \\ 1:5 \\ 1:4 \end{matrix}\right\} \text{ occurs when the crank is } \left\{\begin{matrix} 17 \text{ min.} \\ 34 \text{ min.} \\ 59 \text{ min.} \end{matrix}\right\} \text{ towards } D,$$

from the position when the crank and rod are at right angles; and this position, for all practical ratios of crank to rod, is quite near enough.

The position at which the crank and rod are at right angles can be found graphically thus: from O, in Fig. 175, lay off OA and OB at right angles, and equal to the crank and connecting-rod, respectively, to some convenient scale.

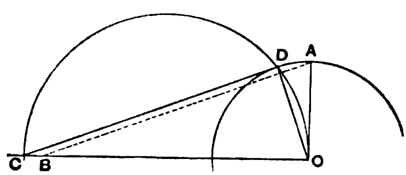


FIG. 175.—CRANK AND ROD.

Lay off $OC = AB$, and on OC, as a diameter, describe the semicircle cutting the crank-pin orbit in D; then OD and CD are at right angles, for the angle CDO is inscribed in the semicircle CDO. The point C, in Fig. 175, corresponds with the point q' in Fig. 174, where the inertia-curve cuts the zero-line of pressure. This point is the same for the forward and the return strokes, neglecting friction of reciprocating parts centrifugal action of rod in its oscillations, and the vertical component of the weight of rod.

The indicator-diagrams shown in Fig. 176 may be regarded as assumed; or they may really be the diagrams taken from the engine, the data for which was previously assumed, for the purpose of investigating the influence of the inertia of the reciprocating parts. The pressure producing motion of the piston is the difference between the forward pressure on one side and the back-pressure on the other side of the piston. That is, Fig. 176, $p1$ and $r2$ represent the unbalanced pressure per square inch on the piston when it is at the positions 1 and 2, respectively. At the right-hand end of the indicator-diagrams the back-pressure for the inboard diagram runs above the forward pressure of the outboard diagram, giving an unbalanced back-pressure represented by $u8$ and $v9$ for the positions of the piston coincident

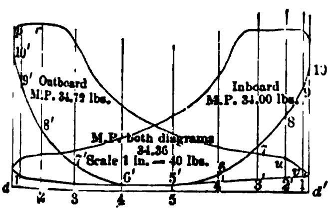


FIG. 176—DIAGRAM.

with these ordinates, respectively. From dd', the zero of pressures in Fig. 177, lay off $p1''$, $r2''$, $u8''$, and $v9''$ equal to the ordinates having same letters in Fig. 176. There results for the outboard end the true diagram *pruv*, the ordinates above the line $d'd$ indicating that the forward pressure is greater, and the ones below that the back-pressure is the greater. A similar diagram is constructed (but not lettered) for the inboard end.

The ordinates of the curve $m'n'o' - v'w'x'$, Fig. 174, are subtracted from the corresponding ordinates of the true diagram *pruv*, Fig. 177, and the ordinates of the curve $x''w''v'' - o''n''m''$ of Fig. 174 are subtracted from the corresponding ordinates of the unlettered diagram for the inboard end, remembering that subtracting a negative ordinate is equivalent to adding a positive one. The subtractions in this operation are performed with the dividers. The inertia-ordinates in Fig. 174 are subtracted from the upper ends of the corresponding ones of the true diagram in Fig. 177 from d to q', after which the ordinates in Fig. 174 becoming negative must be added to the corresponding ordinates of Fig. 177.

The ordinates of the resulting figures give the resultant pressure on the cross-head pin, which is effective in producing rotation of the crank.

The figure of which the cross-section lines incline to the left is the diagram for the outboard, and the one with cross-section lines inclining to the right is the diagram for the inboard end. These cross-sectioned diagrams represent the indicator-diagrams corrected for inertia of the reciprocating parts. For convenience of reference these cross-sectioned diagrams will be termed *corrected diagrams*. The ordinates are to the same scale as the original indicator-diagrams—that is, 1 inch = 40 lbs.

The initial pressure on the piston, the point of cut-off, and the amount of compression can usually be adjusted so that the tangential pressure will be near enough constant to insure smooth running of the engine. Compression, such as shown by the horizontally cross-sectioned part of the corrected dia-

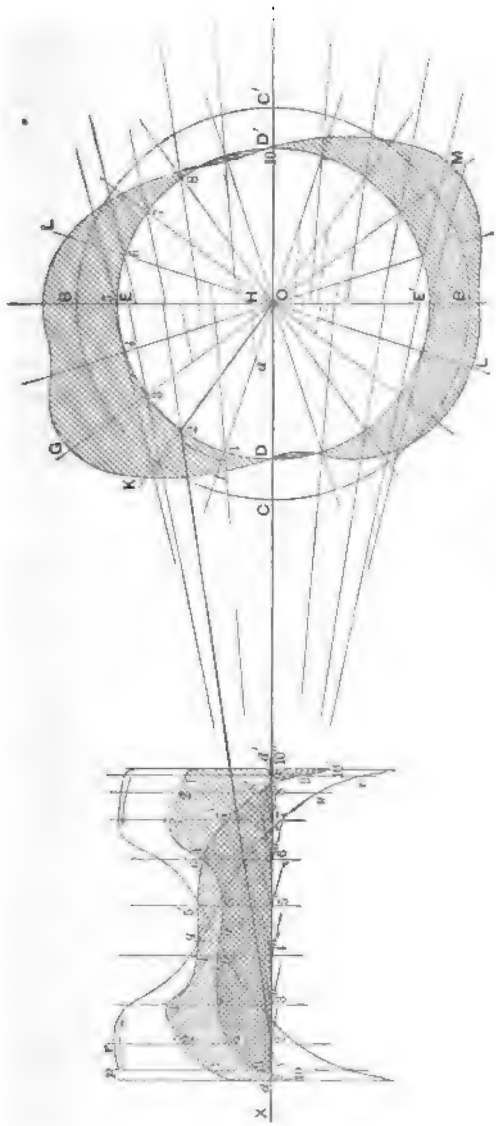


FIG. 177.—TANGENTIAL PRESSURES ON PIN.

gram in Fig. 177 is carried too far. There should be no part of the corrected diagram below the zero-line *d'd*.

There yet remains to be shown graphically the influence of the angularity of the rod on the effective piston-pressure in its transmission to tangential pressure on the pin. Equation (5) shows how the tangential pressure on the crank-pin can be derived from the piston-pressure. If, in Fig. 173, *OB* be drawn to represent to scale the pressure on the piston, *OH* will represent to same scale the tangential pressure on the crank-pin when the connecting-rod is in the position *AB*.

Take the ordinates of the corrected diagrams and lay them off from *O* on the corresponding positions of the crank, and through the outer ends of these radial ordinates draw a line parallel to the corresponding position of the connecting-rod: the intercepts on the vertical radius *OE* will give to scale the tangential pressure on the crank-pin. In illustration of the foregoing, *Oa* is laid off equal to 1'1'' in the corrected diagram; through *a* draw *aH* parallel to the connecting-rod 1'1''; then *OH* will represent to scale the tangential pressure when the crank-pin is at 1, and so on for the other ordinates. The pressure which *OH* represents to scale is the total tangential pressure on the crank-pin at the position 1, divided by the area of the piston. From these tangential pressures as ordinates may be constructed a tangential diagram, either on a straight or a circular base. For an operation occurring in cycles the latter construction seems more appropriate. The crank-pin circle may be taken for the base of the ordinates, and from it lay off $1t = OH$, and so on for the other positions. Joining the outer ends of these radial ordinates gives the tangential diagram *DtGLD'ML'*. The tangential pressure for any point can be found by drawing a radial line through the point and measuring with the scale (1 in. = 40 lbs. in this case) the part of the line between the base-circle and the curve of the tangential diagram.

The twisting moment is found by multiplying the tangential pressure at any point by the length of the crank; but as the length of the crank is a constant factor, the relation of the

ordinates of the tangential diagram will be the same as that of the ordinates of the diagram of twisting moments. And for purposes of comparison the former diagram is all that is required.

The points where the corrected diagrams cut the zero-line of pressures give the corresponding points where the curve of the tangential diagram passes inside the base-circle; showing that the crank-pin is dragging the reciprocating parts.

The curve of tangential pressure will always pass through the points D and D'.

The distance CD represents the mean tangential pressure for one revolution, found by taking the mean unbalanced pressure from the diagrams in Fig. 176, and multiplying it by twice the stroke; this gives the work on the piston for one revolution, and must equal the work on the crank-pin for one revolution. Equating these quantities gives

$$P^m \times 2 \times 3 = P^{mt} \times 3 \times \pi. \quad . \quad . \quad . \quad . \quad (13)$$

$P^m = 33.65$ lbs. = mean unbalanced pressure of both diagrams of Fig. 176.

$P^{mt} = 21.42$ lbs. = mean tangential pressure on the crank-pin = CD to scale. The circle $CBC'B'$ shows at a glance when the tangential pressure is equal to, greater, or less than the mean tangential pressure for one revolution, and how much the former varies from the latter.

Were this engine used for driving a vessel, the propeller should be set on the shaft so that the blades would not pass the rudder-post when the tangential pressure on the crank-pin is at its maximum.

In practice, one figure may take the place of Figs. 174 and 177, which are here used to avoid a confusion of lines in the explanation of the method. Many lines may be omitted; as two points of a line are enough for its direction, and this is all that is required in many cases.

The original indicator-diagrams from an engine may be used directly by taking the half-length for the length of the crank, which must be divided into as many equal parts as there

are feet in the velocity of the crank-pin per second, and then multiplying P', equation (9), by the number of pounds found by the indicator-scale in one of these divisions of the crank-radius.

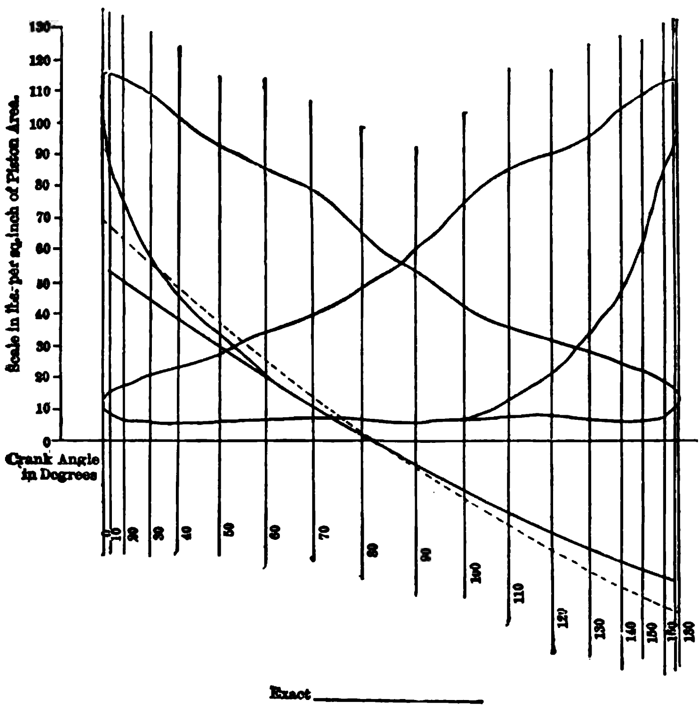


FIG. 178.—LOCOMOTIVE ENGINE.

The pressure due to the inertia-stresses of reciprocating parts being, approximately,

$$p = \frac{Wv^2}{gr}(\cos\theta \pm m\cos 2\theta),$$

—where W is the weight of those parts per square inch of piston, and θ is the angle of rotation, measured from the beginning of stroke or its end, and v and r, as usual, the velocity of

rotation and radius of crank—the inertia-curves may be constructed, and being read to any suitable scale, will answer for any engine, whatever the values of v and of r.

118. Modifications of the Action of the Engine have been carefully studied by Professor Jacobus, who has obtained

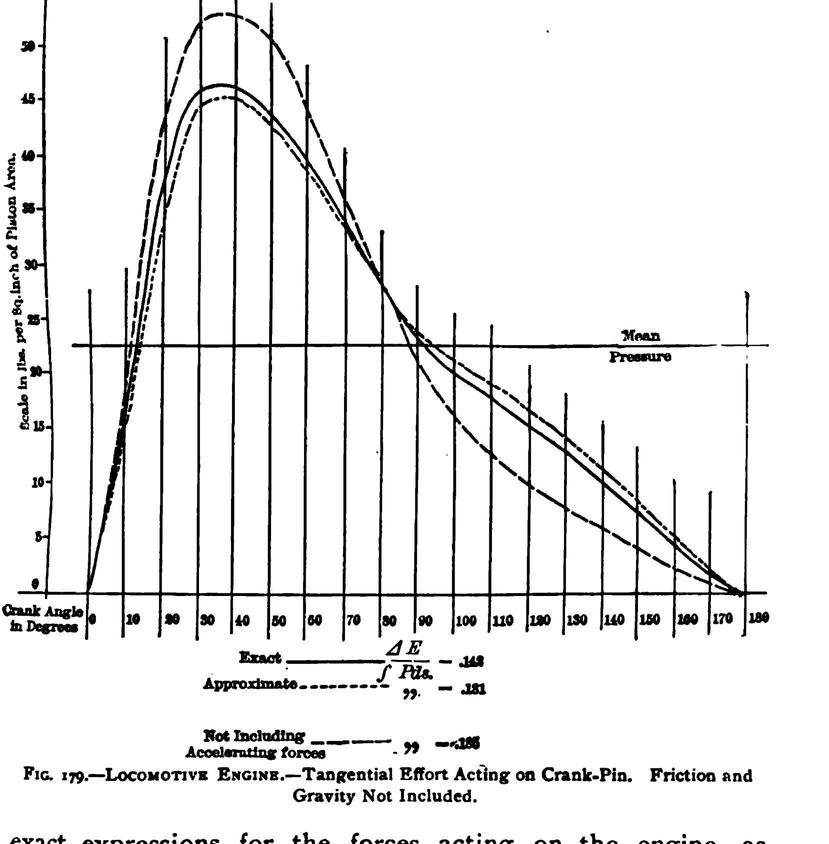


FIG. 179.—LOCOMOTIVE ENGINE.—Tangential Effort Acting on Crank-Pin. Friction and Gravity Not Included.

exact expressions for the forces acting on the engine, as influenced by friction, acceleration, and gravity.* The conclusion is reached that the usual approximate methods may be

*Trans. Am. Soc. Mech. Engineers, vol. XI.; N.Y. Meeting, 1889; No. CCLXXVII.

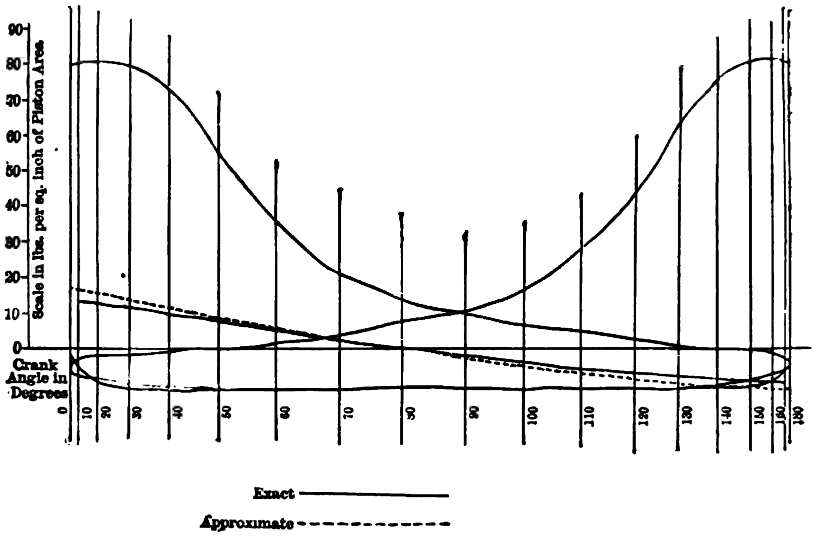


FIG. 180.—CORLISS ENGINE.

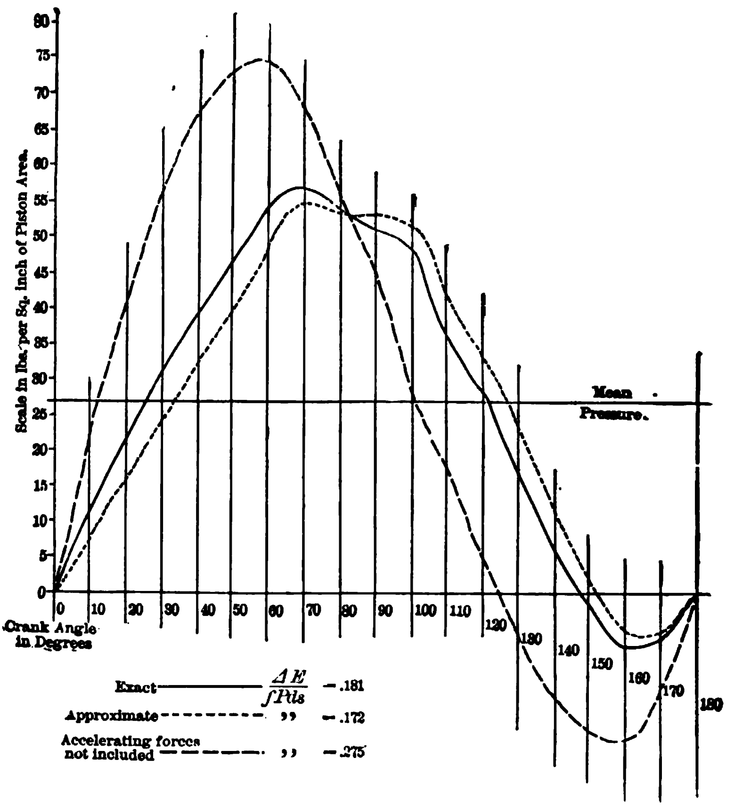


FIG. 181.—CORLISS ENGINE.—Tangential Effort Acting on Crank-Pin. Friction and Gravity not Included.

safely accepted for all the purposes of the designing engineer. Figs. 178–181 exhibit the resultant forces acting on engines having the following dimensions:

	Class of Engine.	
	Large horizontal high-speed. Locomotive.	Large horizontal slow-speed of revolution. Harris—Corliss
Revolutions of crank-shaft per minute	250	60
Length of stroke, in inches	24	60
Diameter of cylinder, in inches	18½	26½
Length of connecting-rod, in inches	92	150
Distance from the wrist-pin to the centre of gravity of rod, in inches	55	66
Distance from the centre of the crank-shaft to the line of travel of the wrist-pin, in inches	0	0
Principal radius of gyration, in inches	34.1	48
Weight of piston, piston-rod, and cross-head, in pounds	474	1300
Weight of connecting-rod, in pounds	307	1200

To determine an adjustment of weights, take p_x as the intensity of pressure of steam at any point in the stroke x,

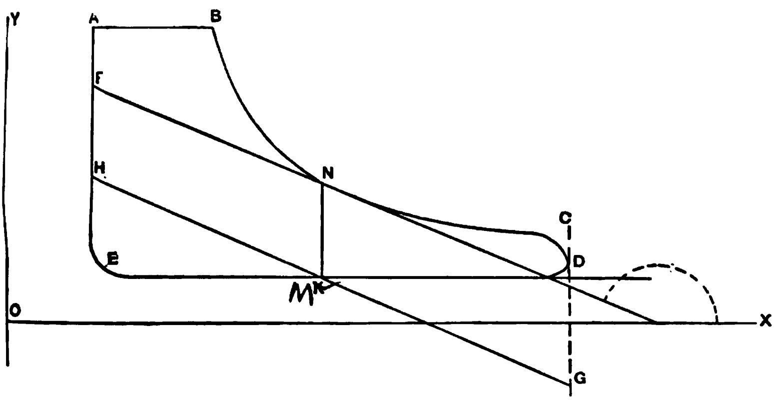


FIG. 182.—ADJUSTING FORCES.

and we have $\tan\theta = \dfrac{dp_x}{dx}$ in the next figure, when θ is the angle made by the line of pressures with the path of the piston.

In order that the net amount of work done in the cylinder shall be nearly equal on both sides the half-stroke position of

piston, draw a vertical MN to cut the expansion-line at N, and draw the tangent DF to that point; then a speed and weight of parts such that the line of acceleration-pressures GH shall be parallel to FD will give, according to Schmidt, very nearly the best adjustment.

Let $S =$ stroke;
$a =$ angular velocity of crank;
$v =$ volume of cylinder;
$V =$ speed of piston;
$c =$ clearance fraction $= \dfrac{\text{clearance-vol.}}{\text{vol. cyl.}}$;
$p_0 =$ value of p_x at half-stroke;
$p_t =$ " " " " end of stroke;

and assuming hyperbolic expansion so that $xp_x =$ const., we have, with Schmidt,*

$$p_0 = p_t \frac{v + cv}{\frac{v}{2} + cv} = p_t \frac{S + cS}{\frac{S}{2} + cS}, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\frac{dp_x}{dx} = \tan\theta = -\frac{p_0}{x_0} = -p_t \frac{S + cS}{\frac{S}{2} + cS} \div \left(\frac{S}{2} + cS\right). \quad . \quad (2)$$

But from what has preceded we have

$$\tan\theta = \frac{p_0}{x_0} = \frac{W}{g} \cdot \frac{a^2R(\cos\phi = 1)}{AR} = \frac{W}{Ag} \frac{a^2R^2\frac{2}{S}}{\frac{S}{2}} = 4\frac{W}{Ag}\frac{a^2R^2}{S^2}; \quad (3)$$

* Dingler's Journal, 1880.

$$\therefore \tan\theta = \frac{p_0}{x_0} = \frac{p_0}{\frac{S}{2} + cS} = 4\frac{W}{g}\frac{a^2R^2}{AS^2} = p_t \frac{S + cS}{\left(\frac{S}{2} + cS\right)^2},$$

$$a^2R^2 = \frac{1}{4}\frac{p_0 gAS^2}{W\left(\frac{S}{2} + cS\right)} = \frac{1}{4}V^2\pi^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

when $V =$ velocity of piston.

Again,

$$a^2R^2 = \frac{1}{4}p_t \frac{S + cS}{\left(\frac{S}{2} + cS\right)^2} \cdot \frac{gS^2A}{W} \cdot = \frac{1}{4}V^2\pi^2, \quad . \quad . \quad . \quad (5)$$

$$V = \sqrt{\frac{p_0 AgS}{W\pi^2\left(\frac{1}{2} + c\right)}} = \sqrt{\frac{p_t}{W} \cdot \frac{(1 + c)}{\left(\frac{1}{2} + c\right)} \cdot \frac{AgS}{\pi^2}},$$

$$= \sqrt{S} \,.\, \text{const.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

when the engine is given; or,

$$W = \frac{p_0 AgS}{V^2\pi^2\left(\frac{1}{2} + c\right)} = \frac{p_t}{V^2} \cdot \frac{AgS}{\pi^2} \cdot \frac{1 + c}{\frac{1}{2} + c}, \quad . \quad . \quad . \quad (7)$$

when it is proposed to calculate W for any given value of V and p'.

Thus, make $p_t = 25$ lbs. per in. $= 3600$ lbs. per ft.;
$A = 100$ sq. in. $= 0.7$ sq. ft.;
$g = 32.2$ ft.;
$V = 10$ ft. per sec.;
$S = 4$ ft.;
$c = 0$; $\pi^2 = 10$, nearly;

$$W = \frac{3600 \times 0.7 \times 32.2 \times 4 \times 2}{100 \times 10} = 324.6;$$

or, taking $W = 2000$;
$p_t \times A = 17,500$;
$S = 4$ ft.;
$c = 0$; $\pi^2 = 10$, nearly;

$$V = \sqrt{\frac{17500 \times 4 \times 32.2}{2000 \times 10 \times \frac{1}{2}}} = 15 \text{ feet per sec.} = 900 \text{ per m.};$$

i.e., 112.5 rev. per min.

Or, if $W = 1100$;
$p_t \times A = 5000$;
$S = 1.25$;

$$V = \sqrt{\frac{5000 \times 1.25 \times 32.2}{1100 \times 10 \times \frac{1}{2}}} = 6 \text{ ft. per sec.} = 144 \text{ rev. per m.}$$

For the common adjustment of pressures in non-condensing engines, in which the steam-pressure is not far from 100 lbs. per square inch and the ratio of expansion usually about four, it may be assumed that the maximum speed may be taken to be that at which the difference of stresses due the steam-pressure on the piston and the effort of centrifugal force at the pin at the end of the stroke is equal to the sum of these pressures at the other end.

Then we have, taking p_c and p_1 as the two pressures at the beginning of the stroke,

$$p_1 - p_c = \frac{1}{r}p_1 + p_c$$

$$p_c = \tfrac{1}{2}p_1\left(1 - \frac{1}{r}\right)$$

$$= 0.000341 \frac{N^2RW}{A}$$

$$N = \sqrt{\frac{Ap_1\left(1 - \frac{1}{r}\right)}{0.000682RW}}.$$

Thus, when $A = 100$, $p_1 = 100$, $r = 4$, $R = \frac{1}{2}$, $W = 100$, we shall have a maximum at $N = 500$.

Crank-effort curves may be constructed for any case, once the indicator-diagram, the dimensions of the engine, and the weight of moving parts are known. The diagram is corrected for the inertia-effects of the reciprocating elements, and the pressures applied to the pin then become known. These pressures are analyzed into their radial and tangential components, and the latter give the ordinates for use in the construction of the curves.

Two or more cranks being employed, as in marine engines, these curves may be superpoised on a common base-line, and the total effect thus obtained and graphically registered.

Counter-pressure by cushioning may be readily adjusted to the requirements of any given case, so as to prevent "pounding," by computing the pressure and stored energy to be taken up by compression, at the end of the stroke, and so beginning the compression that that energy shall be just taken up at the moment of approaching the centre.* Its final adjustment is made by trial, after the engine is set at work. In high-speed engines large clearance may be, on this account, necessary; and compression to boiler-pressure for this, as well as for thermodynamic and economical reasons, is often practised.

By a good adjustment of clearance, compression, weight of reciprocating parts, and counterbalance, the easy movement and the safety and durability of very fast-running engines may be assured. Without such adjustment great annoyance and even serious dangers may be incurred. It is evident, however, that no invariably correct adjustment can be prescribed for an engine of varying speed, aside from the counterbalance. Adjustments suitable to its intended regular speed must suffice.

Modifications, small but not negligible, are evidently introduced into the computations of the size of parts of engines by the inertia-forces, and these processes will include the following steps:

---

* For the proper value of this pressure see Chap. VI., § 171.

(1) The intended ideal steam-distribution should be determined, the ideal diagram constructed, and the mean effective pressure obtained.

(2) The weights of reciprocating parts in the design proposed and their inertia-forces should be next determined, and the effect upon the engine represented in its modification of the diagram. Weight should be added, if needed, until this modification is satisfactory at the intended speed of the engine. It is obvious that while the distribution and momentary effects of energy and pressures are effected by inertia, the total effects and total energy expended remain unaffected.

(3) The compression should be adjusted at the end of the stroke, and its effect on the diagram exhibited in its absorption of the energy of the reciprocating parts.

(4) Should this adjustment of compression compel cushioning to boiler-pressure and yet be insufficient, it may sometimes be well to increase the "dead spaces," or clearances, until the work of compression can be made sufficient.

(5) The diagram being finally so constructed as to exhibit the whole cycle of the engine, and all forces affecting it, determine the mean effective pressure and make the final computation of its principal dimensions.

119. Balancing the Engine, a matter of comparatively small importance in pumping, and other slow-moving machines, becomes important in engines of high speed of rotation. It involves, however, a careful study of the various forces acting, and of their internal effects, as well as of their method and extent of action on external masses. As has been sufficiently well shown, the acting of the inertia of reciprocating parts may be beneficial to the interior working of the engine by, to a certain extent, equalizing the forces acting on the crank-pin and giving more uniformity to the action of the fly-wheel and the main shaft. They either reduce the necessary weight of fly-wheel or make the weight adopted more effective. They may also reduce the impacts felt at the shaft, and will usually be of still further advantage in lessening the risks of ineffective lubrication.

These inertia-effects, however, are felt outside the line of connection, and, unless counterbalanced, will produce shake on the foundations and a disturbance of surrounding objects that may prove objectionable, if not dangerous. The jar at the end of stroke, also the "pound" of the engine, is not destroyed by any balance of reciprocating forces; although it may often be lessened by the reduction of the range of reversed forces coming into play on turning the centre.

This "pound" is due to the sudden change of direction of the effort on the pin at the instant that the crank passes the centre-line of the engine, and by the blow thus struck on taking up the "lost motion" in the bearings. The magnitude of this shock is determined by the amount of the newly applied effort, the weight of the parts affected, the extent of the lost motion, and the rigidity of the machine. It can only be reduced by some method that shall insure a slower taking up of this "shock" and a more gradual application of the reversed forces. This can be done by a "negative lead" on the steam side of the piston, and by compression on the opposite side, or by early exhaust-closure and such considerable steam-lead as shall take up the energy of reciprocation before the crank passes the centre, and thus quietly reverse the pressures on the pin. Of these expedients, cushioning, or compression, is that usually resorted to; and without it the construction of "high speed" engines would often involve great risks.

Counterbalancing thus gives smoothness of motion on the foundations, while compression insures smooth operation internally, and a proper adjustment of weights of reciprocating parts has its own separate and peculiar province.

In counterbalancing the reciprocating parts of engines, it is obvious that the proposed location may sometimes influence its direction and extent. Thus, if an engine stands vertically on a firm foundation, it should be well balanced laterally, as the vertical forces can be easily controlled. If, on the other hand, it stands on a floor which is not rigid, it should be counterbalanced in the vertical line, the lateral forces then taking effect on the whole floor and side walls of the building.

An unbalanced engine is in this respect better if vertical in the first case and horizontal in the second.

Counterbalancing, properly, introduces a new mass into the machine, having such weight and velocities that the energies stored and restored in its alternate acceleration and retardation shall, at least approximately, equal those restored and stored in the same succession of instants by the parts of the machine to be counterbalanced. If this can be exactly performed, the result is that the engine might be suspended in mid-air without exhibiting motion due the inertia of its parts. Where, as is usually the case, a mass in rectilinear motion must be balanced by another moving in a circular path, as in a crank or in the locomotive-wheel, the result must be a counterbalance in one line with a corresponding new disturbance in another. Such action is usually taken advantage of to reduce the horizontal movements and jar of the engine, and to transfer the jar to the vertical line, where it can have no serious result if foundations and foundation-bolts are right.

Sickels and Wells have constructed engines in which pistons and connecting parts precisely balance each other by simple duplication of parts with opposite motions, constituting a perfect balanced system. The last-named engineer informs the Author that he has thus built a "double" compound engine with cylinders 18 and 36 inches diameter, 24 inches stroke. At each stroke the main bearings are relieved of 38 tons pressure. The pistons and their connections weigh 8000 pounds, and have a velocity of 1000 feet at 250 revolutions per minute—their regular speed.

Where three cylinders or more act, radially, at equal angles, about one crank-shaft and pin, a balance is secured; and three-cylinder engines set parallel, and operating cranks set 120° apart, give similarly good results.

In Fig. 183 the results of partial and full counterbalancing reciprocating parts by a revolving mass, as in the crank-disk, is exhibited as determined by Jacobus for an engine of 10 inches diameter, 12 inches stroke of piston, making 300

"turns" per minute.* The several curves represent the forces

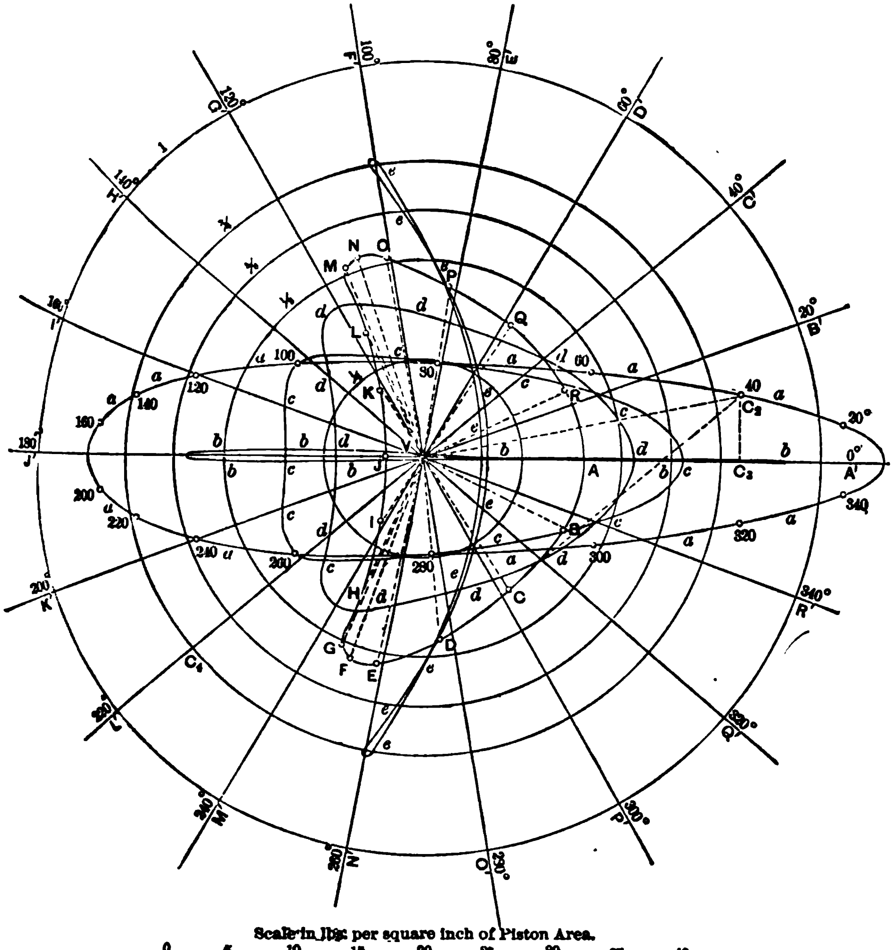


FIG. 183.—COUNTERBALANCING.

of inertia thus acting, when these parts are counterbalanced, thus:

a, *a*, *a*, *a*: curve drawn through the extremities of the lines

* Trans. Am. Soc. M. E., 1889; Fig. 66, vol. XI.

representing the shaking forces, assuming the engine to have no counterweight.

b, b, b, b: curve if a counterweight is employed equal to $\frac{1}{4}$ the mass of the reciprocating parts.

c, c, c, c: curve for counterweight equal to $\frac{1}{2}$ the mass of the reciprocating parts.

d, d, d, d: curve for counterweight equal to $\frac{5}{8}$ the mass of the reciprocating parts.

A, B, C, D, etc.: curve for counterweight equal to $\frac{3}{4}$ the mass of the reciprocating parts.

e, e, e, e: curve for counterweight equal to the entire mass of the reciprocating parts.

The curves thus produced are seen to be in no case precisely elliptical, although they closely approximate that form in some instances. Exact formulas may be employed usefully in computing the forces tending to shake the foundation. The figure shows that the size of the counterweight should be determined, in part at least, by the character of the foundation. The shaking forces should be made principally vertical for a very solid or a tall foundation; in other cases the light weight and horizontal efforts may be demanded.

The most important and troublesome problem of this class is that presented by the locomotive designed for high speeds. The usual solution is, however, only approximate, and is commonly, in effect, that proposed by Clark: *

Find the separate weights in pounds of crank-pin, coupling-rods, and connecting-rod for each wheel; and the reciprocating weight of the piston and appendages, and half the connecting-rod. Divide the reciprocating weight equally between the coupled wheels, and add the part so allotted to the revolving weight on each wheel. The sums obtained are the weights to be balanced at the separate wheels. Multiply by the length of crank, and divide by the radial distance of the centre of gravity of the space to be occupied by the counterweight. The result is the weight of the counterbalance, to be placed opposite the crank, in each wheel.

* Railway Machinery, 1855.

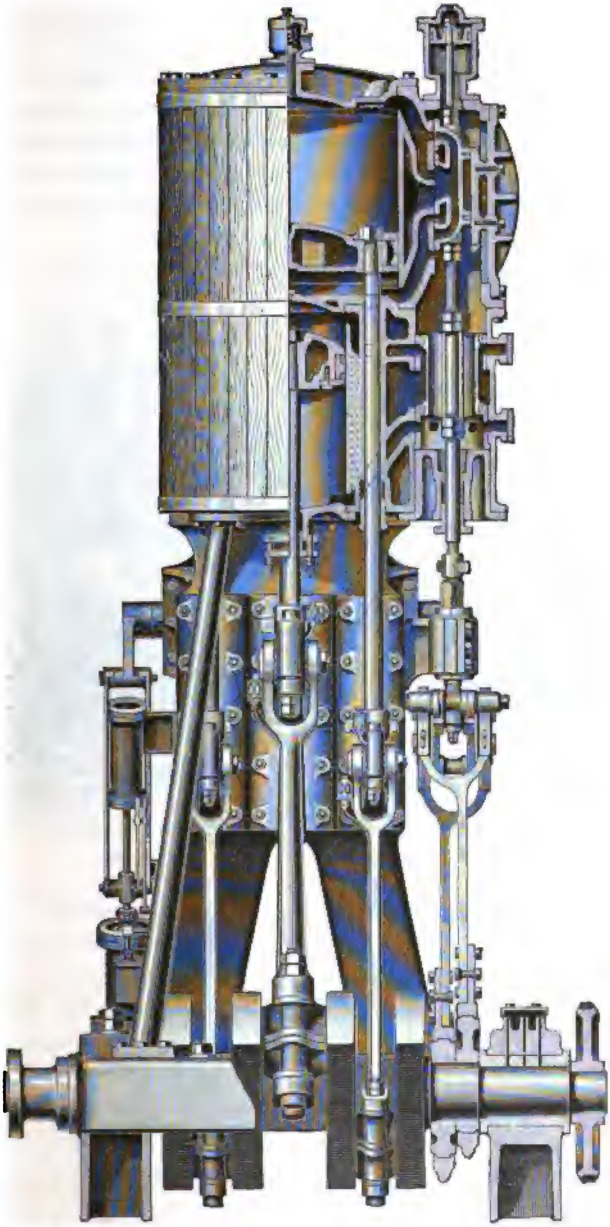


FIG. 184.—WELLS BALANCED ENGINE.

In rare cases, as in Carel's locomotive, the system, above referred to, of Sickels and of Wells is adopted to secure a more perfect balance.* When the balance-weight is placed in the wheel, as is usual, the vertical impulses may become serious at high speed. This "hammer-blow," struck on the rail at each revolution of a driving-wheel, in consequence of the action of the added mass, becomes more serious as the weights and speeds of reciprocating parts increase.

The Woolf type of compound has the special advantage, at

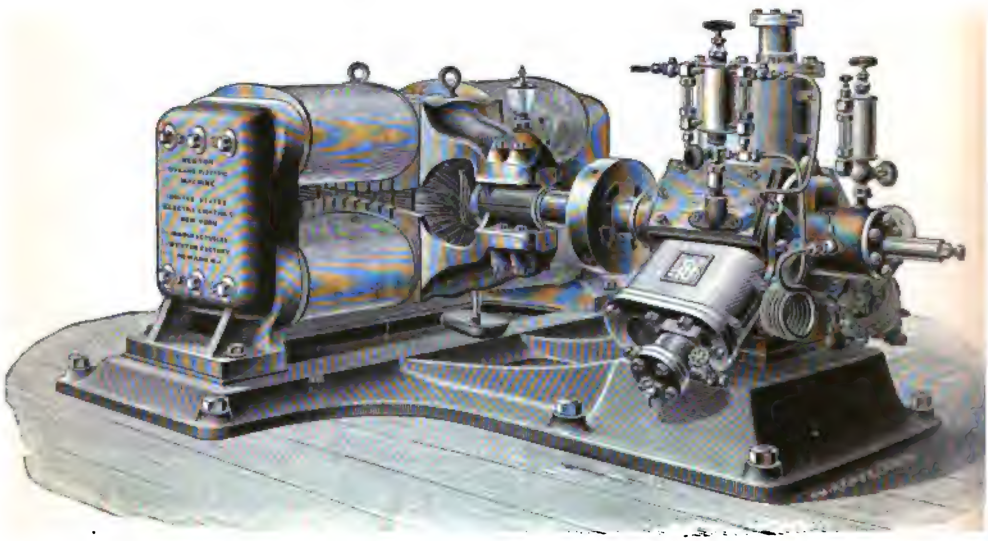


FIG. 185.—THREE-CYLINDER ENGINE AND DYNAMO.

high speeds and with direct connection, of permitting a very thorough balance, with its resulting gain in reduced friction of crank-shaft and smoothness of operation. This fact is well illustrated by the compound engine, designed by Mr. Wells, shown in Fig. 184. Here the weight of running parts is made the same for both engines; the cranks are set opposite, and the balance is practically complete. The work in the two cylinders is also made nearly the same in regular working at

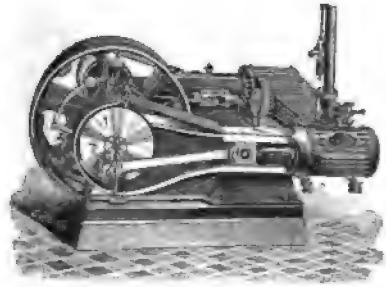
* Vienna Report; R. H. Thurston; 1873.

speed, and the resultant effect is, when these conditions are satisfied, very satisfactory in this respect.

The reduction thus secured in the stresses produced on the frame is worthy of consideration, and the wear of main journals and boxes, in consequence of their relief from load, is likely to prove much less and their life greater.

Quadruple-expansion engines are readily arranged on this principle; although they have no important advantage, in this respect, over the usual three-crank engine.

The Brotherhood type of engine, characterized by its equidistant cylinders—three or four—arranged to connect with a common crank and pin, is an example of complete balancing, and can be driven, like all such engines, up to enormously high speeds. The Author has known them to be carried, experimentally, up to above 2500 revolutions per minute, and 1000 revolutions has been frequently attained. The construction here illustrated is that built in the United States by the Chester Co. This form of engine is illustrated in Fig. 185, as applied to a standard type of dynamo-electric machine. With most forms of even "high-speed" engine, a low-speed dynamo must be especially designed to couple with it; but this class of engine adapts itself to the ordinary speeds of moderate-sized and large dynamos. The steam-turbines demand a special design of high-speed dynamo.



CHAPTER IV.

STEAM-ENGINE CONSTRUCTION, ERECTION.

120. **The Construction, Erection, and Operation** of the steam-engine demand quite a different kind of talent from that requisite to successful design. The determination of the material, the forms and proportions, and the power and probable efficiency of the machinery is the task of the designing engineer, and demands inventive talent, scientific attainments, and a practical knowledge of the circumstances controlling the performance of the machine when at work. The construction of the engine exacts a good knowledge of the quality of the materials to be employed, familiarity with the method and tools of the shop, and experience in the actual work of construction, erection, and operation. The designer must be an engineer in the highest sense of that term; the constructor must be a mechanic of the highest skill, if the work is to illustrate the best results of modern invention and the best work of our time. Each also requires a good knowledge of the work of the other.

The general considerations to be first noted, in this connection, are the character of available material, the nature and efficiency of the tools to be used, and the quality of the labor obtainable. Materials fully in accord with the specification are to be selected, tested for soundness and quality, appropriated each to its special place and purpose, and worked into the required form; the tools best adapted to each operation are to be applied to their various purposes in such manner as to insure that each supplies the largest possible output of finished work of the specified quality; and the workmen are to be given each the kind and quality of work that he can best and most economically accomplish. Every risk is to be

weighed, as well as all specified work considered. It is such careful, exact, and skilful work, with long experience, that has enabled designer and builder, working together, to install two horse-power of machinery per ton of displacement, in modern ships, with a space ranging from 0.35 to 0.45 of their length, and to confine the engineering department of freighting steamers to 0.25 at 12 knots and 0.35 at 18 or 19 knots per hour maximum speed.

The builder should, wherever practicable, check the sizes and proportions, and look for errors in design, and especially in the connection and provisions for access, for inspection and repair, as some mistakes are very likely to arise in the preparation of new designs of any considerable extent and intricacy. He should see that all moving parts are readily and safely accessible when in operation, that removal for repair or replacement can be easily effected, that lubrication is well provided for, noting that allowance of rubbing surface is ample and pressures moderate, the allowable maximum being ordinarily taken at not over 600 pounds per square inch on steadily turning journals, and 1000 on parts like crank-pins, exposed to jar and reversal of pressure. He may check these dimensions by the rules given in Chapter I, Part II. The bevelling, chamfering, grooving, or other arrangement of bearing to insure continuous oil-supply is often left to the constructor.

General considerations determining the character of the materials to be employed in the steam-engine, the methods of working them into shape, of fitting them into place, and of construction and erection, may be enunciated as follows: The work done by the steam-engine involves the application, intermittently and with regular repetition, of powerful forces, and that shake and impact of heavy parts, due to inertia, which is always most trying to mechanism. The magnitude of the forces, useful or detrimental, thus brought into play, compels the use of the strongest available metals; while the impulses and often the impacts which all parts are liable to be called upon to sustain compel the selection of materials of the highest attainable toughness, ductility, and elasticity. Each part

must safely sustain a certain maximum stress, usually calculable, and due to the action of the machine in regular work and to the energy transmitted, or to the surging of heavy parts, or to both combined. These requisites are: (1) strength, and the factor of safety should usually be not less than 6, and is often better 10; (2) elasticity, combined with stiffness; (3) such ductility as will prevent fracture, even if the part be seriously strained. These latter qualities are insisted upon in the highest degree consonant with the first.

The builder endeavors to carefully reproduce the forms and relations of parts prescribed by the designer, and promotes his plans in all practicable ways. He sees that pieces under load have as uniform section as the design permits, especially if of cast-iron, and that castings are free from stress-lines due to variations of thickness; eliminates ribs and brackets on them, when permitted; and anticipates the exigencies and special provisions compelled by the requirement that portions of surfaces must be "finished" or machined. He often is expected to provide for access to every part, and sees proper oil-holes and apparatus of lubrication are properly and amply supplied, and that bolts and nuts are so placed as to permit convenient use of the wrench. He must sometimes determine where brass and bronze shall be employ edin place of iron or steel, and anticipates possible voltaic action. These alloys are not used where liable to serious heating, as liquidation and consequent weakness may result. Fits for piston-rods should be easy, and all such work on the steam-engine looser than in tool-making; and this is especially important where parts are liable to rust fast, as sometimes happens with disastrous effects in the case of badly constructed and mismanaged safety-valves.

The materials of steam-engine construction are mainly cast- and wrought-iron and the various grades of steel; while copper and the alloys find place in the machine to a limited extent. The designing engineer is compelled to assume that the material chosen by him for each part may be obtained of a fair and reasonable quality, such as he may safely base his computations and his proportions upon, and such as he de-

scribes in the specifications which make a part of the builder's contract. In manufacture it is the duty of the builder to see that such materials are selected as are thus demanded, by careful inspection before and during their introduction into the engine, and to thus insure, in this respect, a finally satisfactory construction. He must be intrusted with a certain freedom of judgment in this selection and inspection; but he is required to make his definition of quality conform to that customary in good practice.

In some cases, as, for example, in that in which a certain quality of cast-iron is prescribed which can be best obtained by mixture of two or more grades or brands, the judgment, knowledge, experience, and skill of the constructor must be the main reliance in the attempt to secure safe and satisfactory work. He should therefore be familiar with the materials in the market, their special properties and qualities, and the possible advantages to be gained by their appropriation to specific uses, and by their alloying or mixture, in various proportions, as required for such purposes. A general knowledge of the materials of engineering may be obtained by the study of the special treatises on the subject; but only long experience and a good judgment superadded can give the needed skill in their use. A concise account of the principal materials will be here given.*

The substitution of steel forgings and steel castings for wrought- and cast-iron has effected a great saving in weight, particularly in marine engines; this saving amounting, in some cases in which the substitution has been general, in engines and boilers to 30 or 35 per cent, on weight approximating, in iron, 1000 tons; the weights being thus reduced from about 300 to 200 pounds per horse-power, or the power raised from about 6 to 10 I.H.P. per ton weight of engine and boilers.

The tables in the Appendix may be referred to for data in addition to those to be presented in the text.

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* See Materials of Engineering, 3 vols.; R. H. Thurston. N.Y., J. Wiley & Sons; Revised editions.

Variations of dimensions of castings as they come from the foundry must be anticipated and provided for in the pattern-shop and in moulding. In small pieces these irregularities are commonly due to the "rapping" of the pattern in the sand by the moulder, to enable it to "draw" easily. In large castings this is unimportant; but here the contraction of the mass from the point at which it solidifies and takes the shape and dimensions of the mould may be sufficient, if not anticipated, to cause embarrassment or even actual injury. The "shrink" of plain castings in iron is usually about one per cent, in brass and bronze one and a quarter, and in copper and in zinc one and six tenths and two per cent., respectively. These quantities are commonly expressed as, respectively, one eighth, one sixth, one fifth, and one quarter inch per foot. Steel contracts more than iron. Large masses contract less than small, and intricate castings less than simple forms. "Half-shrink" is often allowed on large pulley fly-wheels and on large gears.

The weights and densities of common metals are given by Fownes as on next page:*

Estimating weights of metal in various forms as used by the engineer is a simple operation. Thus, if

$d =$ diameter of a circular section, or the minor diameter of an ellipse;
$d' =$ major diameter of ellipse;
$l =$ length of piece, section uniform;
$b =$ breadth;
$k =$ a constant;
$W =$ total weight;—

the weight of any piece of uniform section is

$$W = kd^2l \text{ for cylindrical bars;}$$
$$= kdd'l \text{ for elliptical sections;}$$
$$= kbdl \text{ for rectangular sections.}$$

* Chemistry, 10th Ed., p. 297.

WEIGHTS AND DENSITIES OF COMMERCIAL METALS.

NAME.	S. G.	Lbs. in Cu. Ft.	Kilogs. in Cu. M.
Aluminium, cast	2.56	160	2,560
" sheet	2.67	167	2,670
Antimony, cast	6.7	418	6.700
Bismuth, "	9.8	614	9,800
Brass, cast	8.4	525	8,400
" sheet	8.5	532	8,500
" wire	8.54	533	8,540
Bronze (ordinary)	8.4	524	8,400
Copper, bolts	8.85	548	8,850
" cast	8.60	537	8,600
" sheet	8.88	549	8,800
" wire	8.88	550	8,800
Gold, hammered	19.4	1,205	19,400
" standard	17.65	1,103	17,650
Gun metal (bronze)	8.153	510	8,153
Iron, cast, from	6.955	435	6,955
" " to	7.295	456	7,295
" " average	7.125	445	7,125
" wrought, from	7.560	473	7,560
" " to	7.800	488	7,800
" " average	7.680	480	7,680
Lead, cast	11.352	710	11,352
" sheet	11.4	712	11,400
Mercury, fluid	13.6	848	13,600
" solid	15.632	977	15,632
Nickel, cast	7.807	488	7,807
Pewter	11.600	725	11,600
Platinum, mass	19.550	1,219	19,500
" sheet	20.337	1,271	20,337
Silver, mass	10.5	655	10.500
" standard	10.534	658	10,534
Steel, hard	7.82	496	7,820
" soft	7.834	491	7,834
Tin, cast	7.3	456	7,300
Type metal, cast	10.450	653	10,450
Zinc, cast	7.03	439	7,030
" sheet	7.29	456	7,290

The values of k when l is in feet, other dimensions in inches, and W in pounds, are

VALUES OF k

	$W = kdd'l.$	$W = kbdl.$
Brass, sheet	2.906	3.700
Iron, wrought	2.618	3.333
Lead, sheet	3.888	4.950
Steel, soft	2.670	3.400

For pipes, $W = k(d_2^2 - d_1^2)$ when $d_1 d_2$ represent the inner and outside diameters in inches.

To obtain weights in kilogrammes when measures are in centimetres, multiply the above by 0.00241.

121. **Cast-iron** employed in engine-building has a very wide range of quality, and it may be obtained hard or soft, strong or weak, brittle or ductile, as may be required.

For parts of large section, in which weight rather than tenacity are demanded, as in the engine bed or frame, and usually in fly-wheels, an iron is selected easy to work, and which presents a smooth and fair surface, and is of minimum cost consistent with soundness. For parts requiring, as in some cross-heads of stationary engines, great strength and ductility—i.e., large resilience—may be required; in which case the special grades of car-wheel or gun-iron may be specified.

Ordinary grades of good cast-iron have a tenacity of about 20,000 pounds per square inch and very slight extension. The finer grades of car-wheel iron range up to fifty per cent higher strength, and have a very observable ductility. Gun-iron, which is a mixture of the best cast-irons, melted in the foundry air-furnace instead of in the cupola, may attain 35,000 pounds tenacity, and seldom falls under 28,000, while still retaining some extensibility.

Of the cast-irons, "No. 1 Foundry Iron" is the softest grade, is richest in carbon, and is the darkest in color of all the irons; it is weak, moderately tough, of low density, and is quite fluid when molten. It is used principally for mixing with harder grades or for purposes which compel repeated or prolonged fusion. It is the most expensive of all grades of cast-iron. It is to a very slight extent malleable, ductile, and somewhat flexible; is very easily worked by the file and by cutting tools. Its fracture is bright and granular, and of a bluish-gray color. Its texture is finer and more close-grained as its color is lighter. When melted it has much more fluidity than the lighter grades, flows smoothly, and fills the moulds well, taking a good impression from the minutest lines of the mould, and rarely causes trouble by "cold-shuts" or "blow-holes." The

best qualities have a medium fineness and closeness of texture; a clear, dark-gray color; a clean, brilliant fracture, with sharp edges; and a density that is not far from 7.2. Coarseness of grain, a dull color, and irregular structure indicate an inferior iron. When annealed, gray cast-iron is softened, weakened, and reduced in density.

"No. 2 Foundry Iron" contains less carbon than No. 1, is harder, stronger, and denser, and has a finer, closer grain; it is of more frequent use than either of the other foundry grades, and is the iron most called for in all ordinary kinds of work. Its specific gravity is about 7.3.

"No. 3 Foundry Iron" is still lower in carbon, is whiter, stronger, denser, and finer in grain. It is too hard and brittle for general use, and is purchased to mix with softer irons. It often has a slightly mottled surface of fracture.

The specific gravity of the best gun-irons, according to Major Wade, averaged 7.2, and the best of all about 7.26, while condemned guns generally had a density of 7.02 to 7.08. The tenacities of the two classes averaged about 27,200 to 22,100 pounds per square inch (1905 to 1550 kilogrammes per square centimetre).

The following are examples of one specimen taken from each class as representative, respectively, of good and of bad metal for ordnance, or for any purpose demanding, above all other requisites, strength and shock-resisting power, while yet requiring such ease of working as shall make its manufacture reasonably inexpensive: *

	No. 1. Per cent.	No. 2. Per cent.
Carbon, combined	1.70	0.80
" graphitic	2.20	3.20
Silicon	0.30	1.08
Sulphur	....	0.04
Phosphorus	0.44	0.76
Manganese	3.55	1.30

* Wade's Report, Metals for Cannon, pp. 375–396.

Magnesium	0.06	0.10
Calcium	trace	trace
Aluminum	0.28	0.16
Iron	91.84	93.17
Specific gravity	7.22	7.08
Tenacity, lbs. per sq. in.	31,734	18,335
" kgs. per sq. cm.	2,220	1,285

In the above examples the quantity of manganese is remarkably large; in the other examples exhibiting similar differences, it was present only to the amount of a fraction of one per cent. Ordnance iron usually carries considerable manganese, however.

Cold and hot blast irons differ considerably in quality, and this difference is so marked and so generally well understood, that the market-prices of iron made by the two methods differ greatly.

The higher temperature of furnaces having a hot blast causes a more complete deoxidation of the ores, and the reduction of elements which are less readily deoxidized at the lower temperatures of cold-blast furnaces. The effect of heating the blast is, therefore, to cause loss of quality by increasing the proportion of deleterious elements reduced, and which combine with the iron, while greatly increasing the yield of the furnace and decreasing the cost of fuel. When the finest quality of iron is demanded, pure ores, fuel free from sulphur and phosphorus, and flux equally pure must be used. Hence "cold-blast charcoal-iron" is demanded in many cases, to the exclusion of other grades.

It has been stated by some writers that the amount of phosphorus is greater in hot- than in cold-blast iron. This is considered by Percy and other chemists, and by experienced furnacemen, a mistake, as it is found that all phosphorus goes into the iron in any case.

Charcoal, coke, and anthracite irons differ in value, for the same reason that cold-blast and hot-blast irons differ. Coal, as mined, usually contains some impurities, and some kinds of

bituminous coal are very seriously contaminated by sulphur. Anthracite, used as fuel in the blast-furnace, while cheap in certain localities, and convenient to handle, and while giving intense heat, has some objectionable qualities; the "anthracite irons" are, therefore, often found to be of unsatisfactory character. Bituminous coals are sometimes used "raw" in the furnace, and the "raw coal-iron" thus made is often very hot-short, in consequence of the presence of an excessive amount of sulphur. To secure immunity from this injury, the bituminous coals are usually coked, and the iron made with coke is usually, if the flux is free from phosphorus, of good quality. Charcoal has the least proportion of injurious elements of all the fuels used in making iron in the blast-furnace, and the charcoal-irons are therefore of better quality, other things being equal, than the other kinds of cast-iron. Charcoal-furnaces are also usually small, as this fuel is too weak to carry a heavy burden, and the temperature attained within them is less likely to become excessive. They are usually supplied with a blast that is either cold or very moderately warmed—a circumstance which aids in securing excellence of quality of product.

Foundry-work involves the preparation of the moulds, the selection, melting, and pouring of the iron, and the cleaning and delivery of the castings. The mould is commonly made by imbedding a "pattern," or copy of the piece to be made, in sand, thus forming a cavity which, receiving the molten iron, gives the solidifying mass the desired shape. The pattern is often necessarily of complicated form and expensive to make, in consequence of the difficulty met with in giving it such shape, and so separating it and parting the mould that it will easily and safely "draw," leaving the impression in the sand correct in form and uninjured. It is therefore advisable to avoid the use of patterns where the piece to be made is of such form that it may be "swept up" by the moulder without the use of a pattern. Circular sections, and especially heavy parts, are very frequently so made. Screw-propellers, for example,

formerly made from patterns, are now usually formed by the process of "sweeping up."

The choice of iron demands good judgment, large experience, and familiarity with the requirements, in this respect, of the various kinds of castings to be made. Those whieh must be worked or finished by machine-tools must be made of mixtures that can be easily cut and finished. In some cases, as when steam-cylinders are to be cast, it requires much care and judgment to secure a mixture hard enough to be durable under the abrasive action of the piston-rings, and yet not too hard to be bored and faced off. Scrap-iron is commonly employed for hardening, No. 1 foundry for softening, and car-wheel irons for special strength; while No. 3 iron represents the usual machinery grade. The quality of the iron is often much affected by the process of melting in the foundry cupola, and economical work with suitable mixtures depends greatly on the skill of the melter in securing the right mixture, and in obtaining suitable intensity of blast, and a good distribution and intermixture of the charge. Where unusual strength is demanded, it is customary to melt in the reverberatory or air furnace, as in making cast-iron ordnance.*

122. **Wrought-iron** employed in steam-engine construction must generally be of the best quality, in the sense of being sound and homogeneous. It should have a tenacity exceeding 50,000 pounds per square inch, and great ductility. Since the parts composed of this material are those which are exposed to the most serious and dangerous stresses, accidental as well as ordinary, a combination of high tenacity with great ductility is especially demanded.

The following analyses made by Blair,† of four samples of the purest irons supplied in the market for chain-iron, are of value as illustrative of the character of the best weld-irons made for general use, and of very good iron (two each).

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* For an account of the methods of foundry-work, see West's Foundry Practice; N. Y., J. Wiley & Sons, 1888.

† Report of U. S. Board on Tests of Iron, Steel, etc., pp. 247, 248.

Mark.	*O*, $1\frac{1}{8}''$	*L*, $1\frac{1}{4}''$	*Px*, $1\frac{1}{4}''$	*K*, $1\frac{1}{4}''$	*D*, $1\frac{1}{4}''$
Carbon, combined	0.033	0.429	0.057	0.069 }	0.024
Carbon, graphitic........	0.009	0.024	0.009	0.010 }	
Silicon..................	0.073	0.105	0.020	0.159	0.108
Sulphur..................	0.074	trace.	0.001	0.004	0.005
Phosphorus...............	0.078	0.065	0.075	0.161	0.158
Manganese................	0.005	0.006	0.009	0.026	0.038
Copper...................	0.046	0.008	0.008	0.079	0.018
Cobalt...................	0.034	trace.	0.020	0.027	0.031
Nickel...................	0.037	0.011	0.023	0.034	0.021
Slag.....................	0.974	0.326	1.214	0.470	.....

Of irons of lowest grade analyzed are the following:

It was concluded from the study of the series of irons from which the above are selected, that phosphorus may be allowed in any proportion less than 0.10 per cent, and may even be of advantage where the carbon is below 0.03, and silicon less than 0.15 per cent. Silicon in excess, it is concluded, reduces strength; but loss of tenacity and ductility are oftener due to the presence of silica than to that of silicon. It was found that the same brand of iron may vary greatly in chemical constitution and physical character. With iron of fairly good composition, the quality is determined much more by differences in working, and peculiarities in method of manufacture, than by minute differences of composition such as usually exist.

The work of the Committee on Chain Cables of the United States Board appointed to test Iron, Steel, etc., has revealed many valuable facts, and their conclusions may be stated in brief:

"(1) That any wrought-iron, of whatever ordinary composition, may be welded to itself in an oxidizing atmosphere at a certain temperature, which may differ very largely from that one which is vaguely known as a 'welding heat.'

"(2) That in a non-oxidizing atmosphere, heterogeneous irons, however impure, may be soundly welded at indefinitely high temperatures.

"(3) Irons in which there is much carbon require to be welded at a very low heat; phosphorus in excess calls for the

same. Coarse iron with much slag requires a high heat and hard hammering, and even then there is a liability for 'faces' to form throughout the whole surface of the lap, which faces simply stick together, and are liable to draw. A very close, fibrous iron also requires a high heat and hard, rapid hammering; the reason for which is that the heavy blows previously required to make the scarf or lap have so amalgamated the fibres one with another that when the two laps are brought into contact the fibres of each do not intermingle thoroughly, and they, too, are frequently simply stuck together, adhesion taking the place of a process similar to felting, which occurs in welding an iron with a rather open fibre. With this a low heat is required, which seems to penetrate and expand the fibres so that they intermingle, and the two laps are held together by a net-work. Moderate hammering is necessary with this type of iron, which is seldom found to possess great tensile strength, but nearly always has great resilience.

The following conclusions were finally reached: *

(1) Although most of the irons under consideration are much alike in composition, the hardening effects of phosphorus and silicon can be traced, and that of carbon is very obvious. Phosphorus up to 0.10 per cent does not harm, and probably improves, irons containing silicon not above 0.15 and carbon not above 0.03. None of the ingredients, except carbon, in the proportions present, seem to very notably affect welding by ordinary methods.

(2) The strength of wrought-iron and its welding power by ordinary methods are varied more by the amount of its reduction in rolling than by its ordinary differences in composition. Uniform strength may be promoted by uniform reduction, but only at such increased cost of manufacture that the practice is not likely to obtain. Therefore the reduced strength of large bars made by ordinary methods should be considered in designing machinery and structures.

(3) The U. S. Board has demonstrated that the tenacity of

* Ibid., § 188.

2-inch bar-iron, as customarily made for chain-cable, should be between 48,000 and 52,000 pounds per square inch, and of 1-inch bar between 53,000 and 57,000 pounds, and that stronger irons than these make worse cables, because they have low ductility and welding power.

(4) Chemical analyses, made in connection with physical tests, are indispensable to conclusions about either the character or treatment of iron.

(5) Analyses prove that the same brand of wrought-iron may be heterogeneous in composition, and they emphasize the previously known fact that wrought-iron-making processes, as compared with the cheap steel processes, necessarily give an uncertain character to the former material, while to the latter the desired quality may be imparted with certainty and uniformity.

(6) The ordinary practice of welding is capable of radical improvement. The perfection of means for welding in a non-oxidizing atmosphere would seem to be the promising direction of improvement.

Good wrought-iron varies in tenacity from 40,000 pounds per square inch (2812 kilogrammes per square centimetre) upward, and this variation continues, as carbon is added, through the several grades of steel until more than 200,000 pounds per square inch (14,060 kilogrammes per square centimetre) is attained. Cast-iron varies from 10,000 pounds per inch (703 kilogrammes per square centimetre) to above 40,000 (2812 kilogrammes per square centimetre), according to purity and physical character.

Tension members, in heavy structures, when of wrought-iron, are usually calculated for a load of 10,000 pounds per square inch (703 kilogrammes per square centimetre). Struts and compression members are loaded to 8000 (562 kilogrammes), and members subjected to orthogonal stresses are allowed from 5000 to 7000 pounds (350 to 500 kilogrammes). Steel is allowed, in such structures, fifty per cent higher loads; cast-iron is often loaded to one third the given figures.

It is sometimes customary, in bridge-work, to allow an in-

crease of assumed load, when moving, of 15 to 25 per cent on small spans, of 10 per cent on spans of 50 to 100 feet, but no more on large spans.

In some cases it is advisable to design some minor part, or element, of a train with a lower factor of safety, to insure that when a breakdown does occur it shall be certain to take place where it will do least harm. For example, a "breaking-piece" connects the engine-shaft with the rolls in rolling-mills, since cold iron is sure to be entered into the rolls occasionally, and will inevitably break the weakest element of the train transmitting power.

The British "Steel Committee" tested iron and steel by compression in 1868–70, and found the elastic resistance of English wrought-iron to lie between 10 and 14 tons per square inch, averaging about 12, or nearly 26,000 pounds per square inch (1827.8 kilogrammes on the square centimetre), and an extension, to the elastic limit, of 0.097 per cent (0.001 nearly).

Kirkaldy, experimenting on the softer and purer iron of Sweden, obtained an average of about 25,000 pounds (1757.5 kilogrammes on the square centimetre), and an ultimate resistance of nearly 175,000 pounds per square inch, with a 1-inch cube (12,300 kilogrammes on the square centimetre), and about one half that amount on a 1½-inch (3.82 centimetres) bar 2 diameters long. Ten diameters' length reduced the figures to about 15 per cent of the maximum. The compression was nearly 50 per cent.

The *size of section* of the iron is found to seriously affect the strength and other properties. Thus the experiments of the Author give, approximately,

$$T = 56{,}000 - 20{,}000 \log d,$$
$$T_m = 4500 - 1406 \log d_m,$$

for the tenacities of round iron of good quality.

In other cases it was found that

$$T = \frac{60{,}000}{\sqrt{d}},$$

$$T_m = \frac{80{,}000}{\sqrt[4]{d_m}}.$$

The Edgemoor Iron Co. have used the expression

$$T = 52{,}000 - \frac{7000A}{B},$$

in which A and B are the area and the periphery of the section.

The following are standard British tests of iron:

Material.	Ultimate Stress. Tons per square inch.		Contraction per cent of area at fracture.	
	Highest Class.	Lowest Class.	Highest Class.	Lowest Class.
Bars, round and square	27	23	45	20
" flat	26	22	40	16
Angle or T	25	21	30	12
Plate lengthways	24	20	20	8
" crossways	22	17	12	3

Materials tested to 4 per cent of total value.

Materials under specified strain are accepted if the contraction is proportionally higher.

Admiralty tests for iron plate are as follow:

Hot, to bend without fracture from 90° to 125°.

Cold test, to bend without fracture to the following angles:

	Lengthways.	Crossways.
1-in. plate	10° to 15°	5°
¾- " "	20° " 25°	5° to 10°
½- " "	30° " 35°	10° " 15°
¼- " "	55° " 70°	20° " 30°

A common test for cast-iron is 1 ton on the centre of an inch-square bar 1 foot between supports; or 30 cwt. on the centre of a bar 2 inches deep × 1 inch wide and 3 feet between supports—the bar to bear the load without breaking.*

123. Steels used in the steam-engine have as wide a range of quality as of composition. They are of the grades known as machinery steel, and are usually the product of either the Bessemer or the Siemens-Martin, the "open-hearth," process.

* Molesworth.

Steels made by an American firm, noted for the excellence of its "steel boiler-plate," were found to contain:

	A_1	A_2	B_1	B_2	C_1	C_2
Carbon, combined	0.243	0.225	0.375	0.384	0.744	0.733
Carbon, graphitic	0.011	0.019	0.012	0.012	0.012	0.017
Silicon	0.013	0.008	0.070	0.070	0.074	0.067
Sulphur	0.058	0.066	0.038	0.043	0.043	0.042
Phosphorus	0.128	0.132	0.092	0.094	0.104	0.107
Manganese	0.341	0.362	0.685	0.649	0.465	0.471
Copper	0.278	0.308	0.210	0.240	0.346	0.356
Cobalt	0.045	0.047	0.050	0.041	0.052	0.057
Nickel	0.065	0.057	0.115	0.105	0.120	0.135

Of these A may be taken to illustrate the special grades demanded for boiler-plate, in which ductility and non-hardening qualities are essential; B is the common rail and steel also used for parts of machinery; while C is the grade mainly employed where high tenacity is the main requisite. The latter is also an excellent grade for use in steam-hammers, piston-rods, and for similar purposes, involving exposure to continually repeated and violent shock.

The tenacities of such steels have been found, in special experiments made by the Author, to vary nearly as follows:

$$T = 60,000 + 70,000\ C;$$
$$T_m = 4218 + 4921\ C;$$

when T and T_m are the tenacities in pounds on the square inch, and in kilogrammes on the square centimetre; while C is the per cent of carbon present. It is understood that these steels are of good quality.

When annealed, it was found that

$$T = 50,000 + 60,000\ C;$$
$$T_m = 3515 + 4218\ C.$$

The final elongation, per inch or per centimetre, was about

$$El = 6000 \div T;$$
$$El_m = 4.2 \div T_m;$$

and the total resilience, or product of tenacity and ultimate elongation,

$$R = \frac{2}{3} T \times El = 4000 \text{ ft.-lbs.};$$
$$R_m = \frac{2}{3} T_m \times El = 2.81 \text{ kgm.};$$

and nearly constant for all. One square-inch section and one foot length, or one square-centimetre section and one centimetre length, are here taken.

Tests of the strength of ingot iron and steel, especially as affected by variation of composition, were undertaken, 1875–78, by the "U. S. Board testing Iron, Steel, and other Metals." The selected samples of steel then tested by the Author exhibited the widest range of composition, from that of the softest ingot iron to the hardest tool steel. These steels were tested with the result exhibited in the next table.

TENACITY OF INGOT IRONS AND STEELS.*

Carbon. Per cent.	Elongation. Per cent.	Elastic Limit.		Ultimate Strength.		Modulus of Elasticity, *E*.	
		Kilogs. per sq. cm.	Lbs. per sq. in.	Kilogs. per sq. cm.	Lbs. per sq. inch.	Kilogs. per sq. cm.	Lbs. per sq. inch.
.009	29.67	1,683	26,500	3,123	43,000	1,801,859	25,631,000
.057	25.50	2,425	34,500	3,867	55,000	1,566,706	22,286,000
.130	34.33	2,004	28,500	3,656	52,000	1,974,376	28,085,000
.234	20.83	1,828	26,000	4,218	60,000	1,913,288	27,216,000
.238	12.00	3,473	49,400	4,900	69,700	1,727,793	24,576,000
.401	21.67	3,567	50,743	5,012	71,300	1,989,209	28,296,000
.463	20.17	2,840	44,000	4,991	71,000	1,657,127	23,558,000
.577	19.50	3,359	47,800	5,842	83,100	1,847,836	26,285,000
.639	2.75	3,515	50,000	6,643	94,500	1,833,143	26,076,000
.691	3.58	3,550	50,500	7,100	101,000	2,051,987	29,189,000
.756	1.00	4,583	65,190	7,128	101,400	1,616,900	23,000,000
.806	9.75	3,550	50,500	7,902	112,400	2,054,368	29,223,000
.873	8.17	3,550	50,500	7,951	113,100	1,833,143	26,076,000
.923	11.08	3,585	51,000	8,349	118,900	1,805,374	25,681,000
.996	10.08	4,352	61,900	8,591	122,200	1,853,936	26,386,000
1.072	7.67	4,787	68,100	8,647	123,000	1,915,876	27,252,000
1.121	8.08	4,787	68,100	8,823	125,500	1,939,155	27,584,000
1.154	8.67	5,287	75,200	9,166	130,380	1,880,174	26,745,000
1.328	7.33	5,294	75,300	9,412	135,300	1,936,203	27,542,000

* Materials of Engineering, vol. II; Iron and Steel, p. 419 *et seq.*

A singular uniformity of tenacity and of elastic limit is observed within limited ranges of quality, with sudden changes at the limits of each range. On the whole, a gradual increase, both in tenacity and in elastic limit, is seen as the proportion of carbon is increased. The modulus of elasticity varies irregularly within a moderate range, and is evidently not affected by the proportion of carbon present. The quality of the metal is usually determined principally by the proportion of carbon, but is also affected, to a considerable extent, by the silicon and manganese, as well as by phosphorus.

Admiralty tests for steel are as follow:

TENSILE AND EXTENSION TESTS.

(1) Strips cut lengthwise or crosswise of the plate to have an ultimate tensile strength of not less than 26, and not exceeding 30 tons per square inch of section, with an elongation of 20 per cent in a length of 8 inches.

TEMPERING TEST.

(2) Strips cut lengthwise of the plate, $1\frac{1}{2}$ inch wide, heated uniformly to a low cherry-red, and cooled in water of 82° Fahrenheit, must stand bending in a press to a curve of which the inner radius is one and a half times the thickness of the plates tested.

(3) The strips are to be cut in a planing-machine, and are to have the sharp edges taken off.

(4) The ductility of every plate is to be ascertained by the application of one or both of these tests to the shearing, or by bending them cold by the hammer on the contractor's premises, and at his expense.

(5) All plates to be free from lamination and injurious surface defects.

(6) One plate to be taken for testing by tensile, extension, and tempering tests from every invoice, provided the number of plates does not exceed 50. If above that number, one for every addition of 50, or portion of 50. Plates may be received or rejected without a trial of every thickness on the invoice.

(7) The pieces of plate cut out for testings are to be of parallel width from end to end, or for at least 8 inches of length.

When the plates are ordered by thickness, their weight is to be estimated at the rate of 40 lbs. per square foot for plates of 1 inch thick, and in proportion for plates of all other thicknesses: the weight so produced is not to be exceeded, but a latitude of 5 per cent below this will be allowed for rolling in plates of half an inch in thickness and upwards, and 10 per cent in thinner plates.

These weights may be ascertained by weighing as much as 10 tons at a time.

TESTS FOR ANGLE, BULB, OR BAR STEEL.

The whole of the steel to stand a tensile strain of 26 tons to the square inch, and not to exceed 30 tons to the square inch.

Also, to stand the extension and tempering tests described for plates.

All the cross ends to be cut off. One bar is to be taken for testing from every invoice, providing the number of bars does not exceed 50; if above that number, one for every additional 50, or portion of 50.

Lloyd's tests for steel used in shipbuilding are as below:

Strips cut lengthwise or crosswise of the plate, and also angle and bulb steel, to have an ultimate tensile strength of not less than 27, and not exceeding 31 tons per square inch of section, with an elongation corresponding to 20 per cent on a length of 8 inches before fracture.

Strips cut from the plate, angle, or bulb steel to be heated to a low cherry-red, and cooled in water of 82° Fahrenheit, must stand bending double round a curve of which the diameter is not more than three times the thickness of the plates tested.

No reduction will be allowed in the sizes of rivets from those which would be required by the rules for the vessels if built of iron.

In other respects the rules for the construction of iron ships will apply equally to ships built of steel.

"Steel castings" are often so variable in composition and structure as to be safe only when used with a very considerable factor of safety. When well made, however, they are found valuable as a substitute for cast-iron or bronze, as they possess the strength of the best bronze, and cost far less; while their superior strength makes them more desirable, in many cases, than cast-iron, notwithstanding their greater cost. The best "steel castings" have the strength of good forgings, and are therefore used where forgings would be either very difficult to make, or would be too expensive. They should contain less than one half per cent of silicon or of carbon, if sound castings can be secured, and should contain as little manganese as is necessary to give them soundness. Naval work has often been delayed, and even changes of specification compelled, by difficulties in securing reliable steel castings.

It is important that all such castings should be annealed, as otherwise serious loss of strength may be incurred. Steel castings made by the open-hearth method may be obtained of any desired composition, from a minimum of one-fourth or one-third per cent in the hardening elements. They are used for gearing, axle-boxes, cross-heads, and such other forms as cannot be cheaply made of wrought-iron. Their malleability is an important quality, not simply because of the toughness of the casting and its greater safety where shocks are to be met, but also as permitting change of shape by forging.

Copper, rolled and forged, is stronger than castings, and becomes stronger with rolling or drawing.

Major Wade * found the tenacity of Lake Superior *cast* copper to range from 22,000 to nearly 28,000 pounds per square inch (1547 to 1968 kilogs. per sq. cm.), averaging above 24,000 pounds (1705 kilogs.). Egleston gives the tenacity of both Lake Superior and Ore Knob (N. C.) copper as above 30,000 pounds per square inch (2109 kgs. per sq. cm.).

* Metals for Cannon; 1856.

Anderson* gives the figures for the tenacity of copper, which, in round numbers, are as below—ordinary copper is compared with that fluxed with phosphorus:

TENACITY OF COPPER.

	Phos.	Tenacity, *T*.	
		Lbs. per sq. in.	Kilogs. per sq. cm.
Copper, forged	....	34,000	2,390
" cast	....	19,000	1,336
" "	....	25,000	1,758
" forged	0.015	38,000	2,671
" "	0.02	45,000	3,164
" "	0.03	48,000	3,374
" "	0.04	50,000	3,515

The effect of fluxing with phosphorus amounts to an average increase of tenacity of 4000 pounds per square inch (2812 kilogs. per sq. cm.) for each one per cent added up to four per cent.

Sheet copper has formerly been very generally used for the steam and other pipes of marine engines; but its occasional failure in recent high-pressure work has caused serious distrust of brazed copper. "Solid-drawn" tubes and pipe, and, with larger pipe which must be brazed or riveted, reinforcement with steel, iron, or copper wire wrapping, have been used. The substitution of the ductile steels is often considered best.

The shearing resistance of copper is usually given in office hand-books as from 22,000 to 30,000 pounds per square inch (1420 to 2109 kilogs. per sq. cm.). Its value may be taken as the same as in tension, and as subject to the same variations.

The work done in shearing copper is, according to Haswell, measured, for punched holes, by

$$W = 96{,}000dt,$$

in which W is the work in foot-pounds, d the diameter of the hole, and t the thickness of the sheet in inches.

* Strength of Materials.

The copper-tin and copper-zinc alloys furnish a very large number of the best bronzes and engineers' compositions, and are extensively used in every department of construction and the arts. Theywere systematically studied in 1875–8 by the U. S. Government Board upon a plan prepared, proposed, and carried out, at the request of that Board, by the Author, to determine the method of variation of strength, elasticity, and ductility, and of specific gravity, and other properties, with variation of composition throughout all the possible proportions of copper and tin alloys.

The tables given in the appendix exhibit the results of the investigation.*

The tenacities of the valuable class of these metals range not far from 30,000 pounds per square inch (2109 kilogs. per sq. cm.), the strength increasing somewhat with the proportion of tin up to 18 per cent. Within that range the expression

$$T = 30{,}000 + 1000t,$$

in which T is the tenacity and t the percentage of tin, may be taken to represent a maximum which selected materials should give.

Manganese Bronze is a valuable alloy. That used in the construction of torpedo-boats for the British navy was supplied under a contract calling for a tenacity of 26 to 31 tons per square inch (4094 to 4882 kilogs. per sq. cm.), and an elongation of 20 per cent.

This sheet bronze was from $\frac{1}{16}$ to $\frac{1}{8}$ inch (0.16 to 0.32 cm.) thick (No. 9 to No. 18 B. W. G.), and sustained 29 to 30 tons (4567 to 4725 kilogs.), stretching 25 to 35 per cent, and bending cold to a radius equal to their thickness.

The British Admiralty at present (1890) use only phosphor-bronze and gun-bronze in their machinery; the latter having the composition, copper, 87; tin, 8; zinc, 5. Phosphor-bronze is, however, essentially a common bronze slightly alloyed with the phosphorus which has been used as its flux.

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* Consult Materials of Engineering, vol. III, Alloys and their Constituents, by R. H. Thurston; N. Y., J. Wiley & Sons.

The ternary alloys of copper, tin, and zinc were studied by the Author by special methods which need not be here described.* They were made with no other precautions than those observed by every founder, and without using deoxidizing fluxes. The data obtained were consequently quite variable, and the result of this work indicated that the same alloy, especially where the proportion of copper is great, may give very different figures accordingly as it is more or less affected by the many conditions that influence the value of all brass-foundry products.

It was found that the alloys of maximum strength are grouped about a point not far from copper = 55, zinc = 43, tin = 2.

This composition, or something very near it, is the strongest of all, and an alloy of this composition, if exactly proportioned, well melted, perfectly fluxed, and so poured as to produce sound and pure metallic alloy, with such prompt cooling as shall prevent liquation, is the strongest bronze that the engineer can make of these metals. The Author finally made this alloy, and found it a close-grained alloy of rich color, fine surface, and taking a good polish. It oxidizes with difficulty, and the surface then takes on a pleasant shade of statuary bronze-green.

A more minute investigation was carried out, and the alloy made as representing the best for purposes demanding toughness, as well as strength. It contains less tin than the above composition (Cu 55, Sn. 5, Zn 44.5).

Testing the alloy, it was found to have considerable hardness, and tough enough for most purposes. It would forge if handled skilfully and carefully, and not too long or too highly heated, had great strength, and seemed unusually well adapted for general use as a working quality of bronze. In composition it is seen to be a brass, with a small dose of tin.

It had, as tested, a tenacity of 68,900 pounds per square inch (4841 kilogs. per sq. cm.) of original section, and 92,136

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* Materials of Engineering, vol. III, the Alloys; chap. XI, pp. 414, *et seq.*

pounds (6477 kilogs.) on fractured area, and elongated 47 to 51 per cent, with a reduction to from 0.69 to 0.73 of its original diameter.

This alloy was very homogeneous, two tests by tension giving exactly the same figure, 68,900. The fractured surface was in color pinkish yellow, and was dotted with minute crystals of alloy produced by liquation. The shavings produced by the turning tool were curled closely, like those of good iron, and were tough and strong. Fluxing with phosphorus, or the use of phosphor-tin, these alloys were found to considerably increase even these figures.

The results of these investigations showed that bronzes made in the course of every-day business in the foundry should have about the tenacity

$$T_c = 30{,}000 + 1000t;$$

where t is the percentage of tin, and not above 15 per cent. Thus gun-bronze can be given about $30{,}000 + (1000 \times 10) = 40{,}000$ pounds per square inch, if well made. In metric measures

$$T_c^1 = 2109 + 70.3t,$$

giving for good gun-metal $2109 \times 703 = 2812$ kilogs. per sq. cm.

For brass (copper and zinc) the tenacity may be taken as

$$T_z = 30{,}000 + 500z,$$

where the zinc is not above 50 per cent; and

$$T_z^1 = 2109 + 35.15z.$$

Thus copper 70, zinc 30, should have a strength of $30{,}000 + (500 \times 30) = 45{,}000$ pounds per square inch, or $2109 + (35.15 \times 30) = 3165$ kilogrammes per square centimetre.

Referring to the "maximum bronzes" of the Author, a line of valuable alloys may be practically covered by the formula

$$M = z + 3t = \text{Constant} = 55,$$

in which z is the percentage of zinc, and t that of tin. Thus a maximum is found at about $t = 0$, $z = 55$, while the other end of the line is $z = 0$, $t = 18$.

Along this line the strength of any alloy should be at least

$$T_m = 40{,}000 + 500z;$$

$$T_m{}^1 = 2812 + 35.15z.$$

Thus the alloy $z = 1$, $t = 18$ will also contain copper $= 100 - 19 = 81$, and this alloy, Cu 81, Zn 1, Sn 18, should have a tenacity of at least

$$T_m = 40{,}000 + (500 \times 1) = 40{,}500 \text{ lbs. per sq. in.};$$

$$T_m{}^1 = 2812 + (35.15 \times 1) = 2847 \text{ kilogs. per sq. cm.}$$

The alloy Cu 60, Zn 5, Sn 16, should have at least the strength

$$T_m = 40{,}000 + (500 \times 5) = 42{,}500 \text{ lbs. per sq. in.};$$

$$T_m{}^1 = 2812 + (35.15 \times 5) = 2988 \text{ kilogs. per sq. cm.};$$

while the alloy Zn 50, Sn 2, Cu 48, should give, as a minimum per specification,

$$T_m = 40{,}000 + (500 \times 50) = 65{,}000 \text{ lbs. per sq. in.};$$

$$T_m{}^1 = 2812 + (35.15 \times 50) = 4570 \text{ kilogs. per sq. cm.}$$

These are rough working formulas, that, while often departed from in fact, and while purely empirical, may prove of some value in framing specifications. The formula for the value of T_m fails with alloys containing less than 1 *per cent tin*, as the strength then rapidly falls to $t = 0$.

Sterro-metal, a brass containing iron and tin in small proportions, tested at Woolwich, exhibited a tenacity somewhat variable with composition, but always considerable, as seen below.* Its stiffness and resistance to abrasion were also found

* Strength of Materials; Anderson, Lond., 1872.

to be very great. The tenacity may be taken at an average of 60,000 pounds per square inch (4218 kilogs. per sq. cm.), its elastic limit at one half that amount, and its elongation at 0.07. The test-pieces used were three diameters long.

TENACITY OF STERRO-METAL.

Breaking-weight, lbs. per square inch.	Kilogs. per square cm.	Ultimate Elongation at Breaking-point in inches.	Treatment.	Mixture.
46,060	3386	.05	Cast in sand.	Copper 60, zinc 39, iron 3, tin 1.5.
43,120	3032	.015		
54,220	3819	.016	Cast in iron.	Copper 60, zinc 44, iron 4, tin 2.
52,080	3662	.02	Cast in iron and annealed.	
62,720	4410	.045	Forged red-hot.	
60,480	4252	....	After simple fusion.	Copper 55.04, spelter 42.36, iron 1.77, tin .83.
76,160	5355	....	Forged red-hot.	
84,920	5985	....	Drawn cold.	
62,720	4410	....	After simple fusion.	Copper 57.63, spelter 40.22, iron 1.86, tin 0.15.
73,680	5040	....	Forged red-hot.	
82,880	5827	....	Drawn cold and reduced from 100 to 77 transverse sectional area.	

This greater tenacity, as compared with brass and Muntz metal, is probably partly due to the presence of iron, but largely also to the one or two per cent tin. As will be seen later, the Author has obtained higher figures by the use of tin alone.

The use of high-pressure steam and the high temperatures thus introduced make it desirable, in the opinion of many engineers, to secure a piston-packing that will permit dispensing with oil-lubrication. This has been accomplished by the Messrs. Perkins, who use in engines in which the initial pressure is 350 to 400 pounds per square inch a ring composition consisting of copper and tin in the proportions of about three to one.*

* Scientific American Supplement; July 21, 1877; p. 11,283.

Manganese-bronze is now very extensively used, and especially for screw-propeller blades.

The following are considered standard:

ALLOYS USED IN ENGINEERING.

Alloys.	Tin.	Copper.	Zinc.	Antimony.	Lead.	Bismuth.
Brass, engine-bearings	13	112	½	..	..	..
Tough brass, engine-work	15	100	15	..	..	..
" " for heavy bearings	25	160	5	..	..	..
Yellow brass for turning	..	2	1	..	..	..
Flanges to stand brazing	..	32	1	..	1	..
Bell-metal	5	16	..	..	..	..
Babbitt's metal	10	1	..	1	..	..
Brass for locomotive-bearings	7	64	1	..	..	..
" for straps and glands	16	130	1	..	..	..
Muntz's sheating	..	6	4	..	..	..
Metal to expand in cooling	..	..	..	2	9	1
Pewter	100	..	..	17	..	..
Spelter	..	1	1	..	..	..
Statuary bronze	2	90	5	..	2	..
Type-metal from	..	..	..	1	3	..
" " to	..	..	..	1	7	..
Solders.						
For lead	1	..	..	..	1½	..
" tin	1	..	..	..	2	..
" pewter	2	..	..	..	1	..
" brazing (hardest)	..	3	1	..	..	..
" " (hard)	..	1	1	..	..	..
" " (soft)	1	4	3	..	..	..
" " " or	2	..	..	1	..	..

ALLOYS USED IN INSTRUMENTS.

	Tin.	Copper.	Zinc.	Nickel.	Scrap Brass.
"Free" brass	..	2	1	..	..
Common brass	..	1	1	..	1
"Deep-red" gun-metal	1	9	..	..	..
"Best" gun-metal	1	8	..	..	..
German silver, best	..	2	1	1	..
" " 2d quality	..	2	1	.5	..
Babbitt's metal	10	1	..	..	..

FUSIBLE ALLOYS—FOR SAFETY-PLUGS AND FIRE-ALARMS (FUSIBLE AMALGAM).

Melting at 53° C.—Arcet's metal 9 parts, mercury 1 part (Wood's alloy).

Melting at 66°–71°—Lead 2, tin 4, bismuth 7 to 8, cadmium 1 to 2 parts (Arcet's metal).
" " 94° C.—Lead 5, tin 3, bismuth 8 parts.
" " 210° F.—Tin 3, lead 5, bismuth about 8 parts.
" " 246° F.—Tin 4, bismuth 5 parts, lead 1 part.
" " 286° F.—Tin 1, bismuth 1 part.
" " 334° F.—Tin 2 parts, bismuth 1 part, or, tin 3, lead 2 parts.

124. **Modifying Conditions,** as form, proportions, and physical conditions, may greatly alter the usual figures for strength and ductility of iron or steel. It has been seen that wrought iron is less valuable as a material as the size of the parts composed of it increase. In very large sections a tenacity of 35,000 to 40,000 pounds per square inch is considered fair. The same is true of cast-iron and also of steel. All metals are liable to serious depreciation by crystallization with prolonged heating and slow cooling. The defects often noted in large engine-shafts of either iron or steel, the coarseness of grain of large castings in cast-iron or steel, and the brittleness of so-called burned iron are all illustrations of such methods of deterioration, and such as are most generally seen in large masses. The engineer and the constructor, for this reason, allow lower figures for strength and a higher factor of safety on such work.

The forms of various parts determine the magnitude of loads to be sustained by them, or the sizes of the parts for a given loading. Pieces in tension are assigned a section as a minimum, equal to the total load divided by the allowed safe stress. Factors of safety are usually not far from the following:

FACTORS OF SAFETY.

Material.	Load.		Shock.	
	Dead.	Live.		
Wrought-iron; soft steel.	3	6	8 +	Ratio of ultimate strength to working load.
Tool and machinery steel	3	6	9 +	
Cast-iron	4	7	10 to 15	

The proof-strength usually exceeds the working load from 50 per cent, with tough metals, to 200 or 300 per cent where cast-iron is used. It should usually be below the elastic limit of the material, and the value of metals should be generally considered as limited by their elastic resistance.

As this resistance, with brittle materials, is often nearly equal to their ultimate strength, a set of factors of safety, based on the elastic limit, would differ much from those above given for ductile metals, but would be about the same for all brittle materials, thus:

FACTORS OF SAFETY.

Material.	Load.		Shock.	
	Dead.	Live.		
Wrought-iron; soft steel	1	2	3	Ratio of elastic
Machinery steel	1½	3	5	resistance
Tool-steel	2	4	6	to
Cast-iron; foundry	3	6	8 to 12	working load.

The figure given for shock is to be taken as approximate, and used only when it is not practicable to calculate the energy of impact and the resilience of the piece meeting it, and thus to make an exact calculation of proportions.

The factors of safety adopted for iron and steel are lower than those usually admissible for construction in other materials in consequence of the fact that the elastic limit and the elastic resilience, or shock-resisting power of the former, seem to increase, up to a limit, with strain.*

Repeated and reversed stresses are more dangerous than static load; although under the latter the experiments of the Author indicate a danger limit at about 0.6 the ultimate resistance as measured by test.† Wöhler and Spangenberg have investigated the effects of varying stresses and produced formulas to meet the case.‡

* Papers by the Author; Trans. Am. Soc. C. E., 187-379.

† Ibid., p. 591, § 295. ‡ Ibid., p. 622.

Mohr and other writers have shown that the application, above illustrated, of such formulas amounts to the application of a factor of safety of about 3 or 3.5. The cases to meet which this treatment is proposed are fully covered by the factors of safety customarily adopted by engineers.

Wrought-iron in axles is found to have a long life if not strained beyond about 9000 pounds to the square inch (630 kilogrammes per square centimetre).

The *practical proof-strain* is determined by taking one half of the intensity of the stress the piece can sustain a certain maximum number of times without injury. For example, Wöhler found that a rod of Krupp's cast-steel, under a maximum load of 31,132 pounds per square inch (2188 kgs. per sq. cm.), was broken after the load had been applied 45,000,000 times; if this metal had been used in an axle making 30,000 revolutions a day, or 9,000,000 per year, then for five years' duration it might be subjected to a load of 15,566 pounds per square inch (1094 kilogrammes per centimetre).

Vibrations take place on bars loaded to the following limits with equal security against rupture by tearing and crushing:

MAXIMUM LOADS.

	Lbs. per sq. in.	Kilogs. per sq. cm.		Lbs. per sq. in.	Kilogs. per sq. cm.
Good iron, between	16,634,	11,689,	and	− 16,634,	− 11,689
	31,132,	2,188,	"	0	0
	45,734,	3,215,	"	24,941,	1,753
Axle steel, between	29,103,	2,046,	and	− 29,103,	− 2,046
	49,896,	3,507,	"	0	0
	83,113,	5,843,	"	36,386,	2,557
Spring steel, not hardened, between	52,000,	3,655,	and	0	0
	72,736,	5,113,	"	25,520,	1,793
	83,113,	5,843,	"	41,505,	2,921
	93,505,	6,576,	"	62,246,	4,376
For shearing resistance:					
Axle steel, between	22,828,	1,607,	and	− 22,828,	− 1,607
	39,443,	2,772,	"	0	0

For good wrought-iron Wöhler concludes the maximum strain permissible, where the structure is to be permanent, is 8317 pounds per square inch (584 kilogs. per sq. cm.).

Piston-rods, connecting-rods, links, etc., which are subjected

to alternate pull and thrust, should be made about $\frac{8}{5}$ as strong as parts bearing but one kind of stress.

125. The Material for the Steam-cylinder must be carefully chosen, having in view a number of somewhat conflicting conditions: It must be hard enough to wear well under the abrasive action of the piston and its rings; it must nevertheless not be so hard as to make it difficult or seriously expensive to bore out and to "face up." It must be strong and stiff to hold its form under the stresses of varying pressures and temperatures; yet it must not be brittle and liable to crack, even if its jacket be cast solid with it. Such iron can only be made by the experienced founder selecting brands for mixture that may produce the best resultant quality.

In some cases very hard, and even chilling, irons are employed, tempered with some soft, No. 1, iron free from phosphorus. In other instances an admixture of old car-wheels or of new car-wheel iron to ordinary metal, in the proportion of from 15 to 30 per cent, is found to answer well. The larger the cylinder, the harder and the choicer the iron used.

In the American type of steamboat beam-engine, the side-pipes and the valve-chests are cast separately, but are commonly made at the same time and of the same metal as the cylinder itself. Valves are, when of the "double-puppet" variety, still more generally and more scrupulously so made, in order that differences of expansion of the valve and its casing, with changing temperatures and steam-pressures, may not cause leakage.

126. Frames and other heavy parts are commonly made of any good machinery iron, and of a quality which, while fairly strong and tough, is yet easily worked, and will come from the mould with a smooth surface on parts not to be finished. A rather free-flowing iron is most commonly selected. If the casting is very irregular and exhibits abrupt changes of thickness, the danger of shrinkage-cracks will be likely to be considerable, unless care is taken to use iron very "low" in manganese and especially in phosphorus, and to cool very slowly in the flask.

In marine work, frames are often made, in part at least, of wrought-iron or mild steel, in which case it is easy to insure safety by correct proportions.

127. **Rods and Shafts,** and other "running parts," are made exclusively of either the best hammered iron or a low-carbon steel. When of small size no difficulty is found in securing good material or in obtaining safe construction.

In large work, on the other hand, iron is peculiarly liable to be found of irregular and uncertain quality in consequence of the frequent and prolonged heating and cooling in the process of forging and in making numerous welds, and of the production of areas of imperfect welding and volumes of crystallized iron.

Steel is subject to the latter defect also, if either too little worked after leaving the ingot mould, or too seriously heated and cooled, or when worked under a hammer so light that its effect cannot be fully felt at the centre-line of the shaft. Compressed steel, such as is made by the processes of Jones and of Whitworth, is less liable to this defect, especially if but little change of form is required. The middle of the shaft is sometimes, in very large work, drilled out, at great expense, to evade this risk and to permit inspection of the most uncertain portion of the construction. Oil-tempered steel shafts are the most perfect outcome of modern ingenuity in this direction.

The uncertainty of large work in steel, and especially where made under a light hammer, is still such that many engineers adhere to the older material and specify hammered iron. Small parts may be given increased stiffness and elasticity by cold-rolling.

The composition of the best crucible steel for connecting-rods and locomotive parallel rods and parts of engines similarly subject to stress and strain, is not far from

C,	0.004;	Mn,	0.005;
P,	0.0006;	Si,	0.003;

and costs (1890–92) about 6 cents per pound in the United

States. Bessemer and "open-hearth" steels are more generally used, at a much lower cost.

128. **Bolts and Nuts** and minor parts are most frequently, almost invariably, made of forged iron. Manufacturers making it a business commonly supply the market with standard sizes of bolts and screws, and the builder of machinery rarely finds it to his interest to make them. Special shapes and dimensions not standard are, however, often met with, and these are necessarily made by the engine-builder. Oil-cups are supplied by dealers, and are made very cheaply, and in endless variety, of various grades of brass and "compositions" of similar nature.

Straps, gibs and keys, or "cotters," and other small forged parts, are made of the same material as the larger pieces of which they form part. They should usually be of carefully selected material and well made, as the risks of injury are greater than with larger masses. In designing such parts large factors of safety are advisable.

Bolts and nuts and rivets are, unless of very large dimensions, always safely purchased, and will be found true to size and standard, and made of good material. In Europe the Whitworth, and in the United States the Franklin Institute or Sellers, threads are standard for bolts. All invoices should be inspected and reported upon when received, before they are accepted and placed in store. Cold-punched nuts are stronger and cheaper to finish than are hot-pressed nuts.*

The shearing of nuts has been studied by the Author, in an exhaustive series of experiments. It was, however, necessary to turn off the face of the nuts, and to reduce their thickness considerably, in order that they should strip instead of breaking the *steel* bolts on which they were tested. An ill-fitted nut will often strip the thread, and thus fail by shearing the metal; a well-fitted nut will always break its bolt if made of the usual proportions. When turned, as here, they broke

* Materials of Engineering; vol. ii. § 274, p. 521.

through the side, or stripped, with nearly equal frequency. The ½-inch (1.27 centimetres) nuts were 0.44 inch (1.12 centimetres) thick; the ⅝-inch (1.59 centimetres) were 0.6 (1.52 centimetres); the ¾-inch (1.91 centimetres), and the ⅞-inch (2.22 centimetres) nuts were both 0.72 inch (1.83 centimetres) thick.

Nuts broke and stripped without regularity at all loads, and the average figures given may be safely assumed to be fair figures for the basis of calculation of strength on the assumption that the nut will rupture by shearing.

It would probably be safe to take the shearing resistance of cold-punched nuts at 40,000, and of hot pressed nuts at 35,000, pounds per square inch (2,812 and 2,460 kilogrammes on the square centimetre) in ordinary work.

The strength of iron and steel is often greatly affected by punching, and it is usually specified, for riveted work, that no steel plates shall be punched to size, but that all rivet-holes shall either be punched small and the holes reamed out to size, or, better, that they shall be drilled and their edges slightly chamfered or rounded. With soft ingot-iron, such as only should be used for steam-boilers, this injury probably does not occur. Hard iron is sometimes injured by punching, but soft iron may even be strengthened by the process, sometimes as much as 10 per cent.

Pins and Bolts in shear should be calculated with a large factor of safety when in thin metal, in order that they may not cut the eye.

A board of U. S. naval officers has recommended the following proportions:

Breadth of bar	1
Diameter of pin	0.917
Thickness each side	0.555
Thickness at crown	0.722
Depth of eye	equal to thickness of bar.

The resistance that may usually be calculated upon, assuming iron used having a shearing resistance and tenacity of 50,000 pounds per square inch, is taken as for riveting, thus:

	Lbs. per sq. in.	Kgs. per sq. cm.
Double-riveted joint..................	35,000	2,460
Single-riveted joint..................	28,000	1,468
Single-riveted joint—breaking-joint...	34,000	2,390

129. **Steam-cylinder Construction** includes the moulding and casting of the rough piece, finishing it to size as shown by the drawing, and fitting to it its accessories, and the attachment of the cylinder itself to the engine-frame.

Where simple in form and of very moderate size, no special difficulties arise in the foundry, and any good moulder can produce a good casting. The large and sometimes intricate forms often demand in marine and water-supply engineering, and especially when, as is often the case in engines of moderate size, the steam-jacket is cast in one piece with the cylinder, exact the highest art of the foundry. The design of the pattern or the devising of the best methods of "sweeping up," like the planning of the centering of a bridge, often requires more thought and greater skill than the main design itself. The pattern must "draw" safely and easily; the cores must stand securely and in precisely the intended positions; the "gates," "vents," and "rising" or "sinking heads," as the latter are variously called, must all be placed and proportioned with sound judgment, with a view to securing rapid inflow of the iron, free exit of air and gases, and a solid casting, while, at the same time, insuring that no part of the fragile sand shall be carried away or defaced, and no core started out of place by the stream of molten iron.

In some cases it is found advisable to make use of dry-sand moulds very freely; and this has been done even in making cylinders, cores and all, when the jacket has formed a part of the casting.

The casting being made, and a sufficient time being allowed for safe cooling, it is removed from the sand, pickled in very dilute acid, if necessary, to remove the superficial silicated crust, and then carefully cleaned out wherever cored, and finally inspected and passed, before it is sent into the shop for finishing. The sand should be removed from the jacket and

other concealed spaces with especial care, both to prevent cutting of the cylinder if any sand should find its way under the piston while at work, and to insure efficiency of the jacket, which is sometimes not nearly as effective as it should be in consequence of partial closure by the incompletely removed core. The clean and perfect casting is finally delivered at the boring-lathe.

The machine-work consists in boring, facing off the flanges and the valve-seats, fitting the heads and bonnets, and finally attaching to the frame. Often, in some small work, the cylinder, the forward head, and the frame are cast in a single piece. In such cases it is not unusual to have a special machine on which the work may as nearly as possible all be done without moving the cylinder, and at the same time boring out the cylinder and the main bearings in perfect alignment. The more completely the cylinder can be finished —bored, faced, and bolt-holes drilled—without expenditure of labor or risk of loss of alignment in moving it, the better.

Where, as sometimes happens, the cylinder is too hard to cut with ordinary tools, special steels or chilled cast-iron may be made to do the work. It is wise to adopt a high speed and heavy cut, to insure minimum cost; and it is occasionally found economical to do so in such a radical manner as to compel the regular use of the chrome, tungsten, or other self-hardening steels, thus attaining speeds exceeding twenty feet per minute on common iron and with very coarse feeds and deep cuts. The finishing cut is made rapidly, with a broad, round-nosed tool.

When the jacketing is done by using a "liner," the latter should be nicely but easily fitted, and the end adjustment "made to drawing" with absolute exactness—a rule, however, which is equally good for all parts of the construction. Faced surfaces, between covers or bonnets and the cylinder, should be so well finished at the last cut of the tool that a paint of red-lead and oil will stanch any leak, if even that be needed at all.

Where the builder is left any discretion in the matter, he

should use head and nut bolts in preference to stud-bolts, wherever practicable.

The preservation of the exact centre-line, through cylinder, stuffing-boxes, guides, and shaft-centre line, and the rectangular adjustment of the two lines, where not absolutely fixed and right, through the construction of the boring-machine, should be made exact, and with the most scrupulous care.

In fitting up the piston and its rings, care is to be taken to see the former brought to size and made a loose fit in the cylinder, and the latter a nice fit in their grooves or under the follower. The latter should be absolutely a steam-tight fit on the piston, and the whole structure should be at once steam-tight and easy of movement either with or without steam.

Rings to be turned taper should be given precisely the taper shown by the designer, as it is usually the result of exact computation. The shop-manager should, however, as in other directions, revise the designer's construction with a view of detecting possibly impractical, costly, or inconvenient forms, proportions, and plans, as seen from his own standpoint.

Engine-cylinders, and, where jackets are fitted, all casings, are always best bored in the position in which the engine is to be worked, if vertical or inclined, and, in the case of horizontal engines, under conditions giving as nearly as possible the form which the parts will tend to assume when in use. Flanges should be turned and faced off when on the boring-mill or lathe, in the same position as when bored, and, if practicable, without removal between these operations.

Large cylinders should always be bored in the position—horizontal or vertical—in which they are to be finally set, in order that very sensible distortion may not occur and cause trouble and leakage. Large cylinders, when cast with jackets, are very liable to crack, and are now either made with expansion-joints, or with the cylinder and jacket cast separately and nicely fitted together. In this case great care is necessary to secure accurate fitting and to insure tight and permanent joints, with entire freedom from strain.

When piston-rods and valve-stems are run through un-

packed openings, their bushings should be made long and nicely fitted, and especial pains should be taken to get them absolutely in line. All rods should have easy fits, however, and must therefore be made of absolutely uniform section, and perfectly straight. Packed rods should be turned to a loose fit, and a play of light around them, through the hole in which the rod works, is indicative of safety against abrasion. Stuffing-boxes for non-metallic packing should be made as deep, and the gland-followers as short, as is practicable. A deep box filled with lightly compressed packing gives the best results.

130. **The Engine-frame,** if integral with the cylinder, is, if practicable, finished with the latter. It rarely in itself presents any difficulties, either in moulding or in pouring in the foundry, or in fitting up in the shop. The special requirements are, in construction, that the casting shall be smooth and sound, and of homogeneous, close-grained, and fairly strong iron; that the fitting up shall leave the two principal centre-lines absolutely correct in location, and that the surfaces on which the attached parts are to be secured shall be precisely as shown on the drawings. A special tool is often designed for use in finishing the frame, without necessity of moving it after it is once bolted on its table.

The frames of marine and other large engines are often made in several parts, and even, in some instances, serve the multiple purpose of engine-supports and frames, guides and condensers.

Making the frame and fitting to it the cylinder and other parts constitutes one of the most important details of the work of steam-engine construction. The first essential is to see that all finished surfaces, to which the smaller parts are to be fitted, are planed, turned, milled, or bored, as the case may be, exactly parallel to the principal centre-lines of the engine, or to such relations with them as the designer of the machine may have intended, as shown by the drawings. The next most important matter is to see that precise dimensions are secured, to within the limits of exact measurement available. In most cases a fit to within one one-hundredth of an inch on

heavy work should be insisted on, and in some cases to within even one one-thousandth of an inch (0.0025 cm.) of the figured dimensions. The seats for the guide-bars, the face receiving the steam-cylinder, and the centre-lines of cylinder and shaft, are principally important in this regard. The work should be done by properly designed machines, and hand-work avoided as much as possible, as a matter both of accuracy and of economy. The more this end is kept in view, and the larger the lots in which the engine and its parts are built, as a rule, the greater the economy of construction.

131. **Guides, Journals,** and other rubbing parts demand smooth as well as otherwise absolutely correct surfaces, and are made of materials having special fitness for such purposes. Where large areas of cast-iron are made the rubbing surfaces, it is preferred that it should be rather porous and coarse-grained. White-metal linings are employed where the speeds and pressures are high; and the journals in such cases are often ground to precise form after they have been given nearly the desired dimensions in the lathe. Small engine-journals are sometimes of steel, hardened in oil and ground.

Where, as in locomotive work, the guides consist of bars or girders supported at their ends, care in selection of good and sound material is especially demanded. Springing sometimes occurs also in the shaft where it overhangs, and is liable to cause heating where the bearing is not stiff, strong, and well backed by the frame, quite out against the crank; which should in such constructions be made with as thin a hub, in the line of the shaft, as considerations of strength will permit. An over-hung crank is evidently most liable to cause trouble, and such an arrangement compels peculiar care on the part of the builder.

Materials in bearings are commonly the class called "anti-friction" metals. The fact that the friction is often dependent rather upon the nature of the lubricant than upon that of bearing or of journal, makes this a matter of minor importance. It is, however, sometimes advisable that the bearing should be made of a material which can accommodate itself to the

surface of the journal, and thus secure a good distribution of pressure; and in such cases the white alloys, usually containing considerable proportions of tin, are used. This practice is especially common in the case of "high-speed engines," and on large marine engines. The conductivity of these alloys is less than that of the gun-bronze, which is the usual substitute for them, and the same amount of friction causes more rapid rise in temperature.

Bronze for bearings and pieces subject to severe friction, as in machinery, is made of many proportions. Gun-bronze is one of the best: the Author has known of one case in which the bronze was made of ingot copper 90, ingot tin 10, and used in the main crank-shaft journal of a steam-vessel for ten years without appreciable wear, although the area was not unusually large for the load and the velocity of rubbing was high, as is usual in screw-engines. The proportions given in several cases will be found elsewhere: they vary in practice from 88 to 96 per cent copper, as more or less hardness is required. Bronze for steam-engine packing-rings is sometimes made of 92 to 94 copper, 7 to 9 parts tin, 1 part zinc.

The use of 8 to 15 per cent of tin and 2 per cent zinc in alloy with copper is probably as common as the employment of the bronzes without zinc; the latter is added to improve the color. Alloys of copper containing from 3 to 8 or 10 per cent zinc and from 8 to 15 per cent tin are used in engineering very extensively, the softer alloys for pump-work, the harder for turned work and for nuts and bearings. An alloy of 5 per cent tin, 5 zinc, and 90 copper is cast into ingots and remelted for general purposes. It is tough, strong, and sound. Copper 75, tin 12, zinc 3, makes a good mixture for heavy journal-bearings.

Bischof and Bolley have collated numerous tables of alloys employed for various purposes, among which are the following having interest in this connection.*

* Materials of Engineering; vol. iii. § 143.

MACHINERY BRONZE.

	Copper.	Tin.
Malleable bronze (Lafond)	98.04	1.96
Eisler's yellow bronze (golden), hard and elastic	94.10	5.90
Gearing	91.30	8.70
Köchlin's alloy for bearings	90.00	10.00
Seraing " " "	86.00	14.00
Carriage-wheel " "	84.00	16.00
Dies work well on the bronze	83.30	16.70

In further illustration, we have the following as the compositions adopted by the Paris, Lyons and Mediterranean Railway of France :*

STANDARD ALLOYS.

Alloy.	Proportions.					Uses.
	Copper.	Tin.	Zinc.	Lead.	Ant.	
Gun-metal, 1...	82	16	2	..	..	Bearings.
" 2....	84	14	2	..	..	Valves, screws, etc.
" 3....	90	8	2	..	..	Cocks, whistles, etc.
Brass, 1....	70	..	30	..	..	Tubes.
" 2....	67	..	33	..	..	Stuffing-boxes, etc.
" 3....	65	..	35	..	..	Handles, latches.
" 4....	63	..	37	..	..	Plates, washers.
White metal,...	5	71	..	..	24	Bearings.
Packing "	..	14	..	76	10	Stuffing-boxes.
Solder..........	..	45	..	55	..	For tin plate.
"	..	40	..	60	..	" zinc "

The best bronze compositions for use in engineering are, according to Guettier,† the following:

For pumps, bolts, and similar pieces:

Copper	88	Copper	90
Tin	12	Tin	10
	100		100

* Ibid., p. 329.

† Guide Pratique; Paris, 1865.

The latter is the softer of the two. Often from 1 to 4 per cent of zinc is added, as already stated.

For eccentric-straps and connecting-rod bearings:

Copper....	83	84	83	84	82	85.25
Tin	15	14	15	14	16	12.75
Zinc.......	2	2	1.5	1.5	2	2
Lead	..	..	0.5	0.5	..	....
	100	100	100.0	100.0	100	100.00

The addition of lead and increase of copper gives softer alloys. Lead is often used more freely than above.

Locomotive driving-axle bearings:

Copper....	74	80	85.25	86	89
Tin........	9.5	18	12.75	14	8
Zinc.......	9.5	2	2.00	..	3
Lead.......	7	..	....	..	..
	100.0	100	100.00	100	100

The Author prefers gun-bronze to either of the above.

For locomotive slide-valves:

Copper phosphate..........................	3.50
Copper..	77.85
Tin	11.00
Zinc..	7.65
	100.00

Connecting-rod brasses:

Copper phosphate..........................	3.5
Copper..	74.5
Tin...	11.0
Zinc..	11.0
	100.0

Axle-boxes:

	No. 1.	No. 2.
Copper phosphate	2.5	1.5
Copper	72.5	73.5
Tin	8.0	8.0
Zinc	17.0	19.0
	100.0	100.0

Parts demanding greater strength:

Copper phosphate	3.5
Copper	85.5
Tin	8.0
Zinc	3.0
	100.0

Zinc is here added to the bronze to aid in securing that homogeneousness which is essentially the result of the addition of phosphorus.

For pistons (rarely needed): copper, 89.75; tin, 2.25; zinc, 8.

For car and locomotive axle-bearings:

Copper	80	79	86	89
Tin	18	18	14	2.5
Zinc	2	2.5	..	8.5
Lead	..	0.5	..	...
	100	100.0	100	100.0

For ordinary stationary machine journal-bearings: copper, 82; tin, 18.

For whistles of locomotives and bells:

Copper	80	81	78	79	78	71
Tin	18	17	20	23	22	26
Antimony	2	2	2	Zinc 6	..	Zinc 1.8 Iron 1.2
	100	100	100	100	100	100.0

The last is the alloy of the famous "silver-bell" of Rouen.

For pump-buckets, valves, and cocks:

Copper	88	88	86.8
Tin	10	10	12.4
Zinc	1.75	2	0.8
Lead	0.25	..	..
	100.00	100	100.0

For hammers (for use on finished work): copper, 98; tin, 2. This alloy will forge like copper; it may be hardened by adding more tin.

For wagon axle-bearings:

Copper	78	Copper	25
Tin	20	Cast-iron	70
Zinc	2	Tin	5
	100		100

The best brasses may be taken, for general purposes, as accepted by good makers, as follows:

For turned work:

Copper	61.7	66.5	74.5	79.5
Zinc	35.3	33.0	25.0	20
Tin	0.5	0.5	0.5	0.5
Lead	2.5	....	....	....
	100.0	100.0	100.0	100.0

The richer colors are given by the higher proportions of copper. The official recipe for work in French dockyards is:

Copper	65.80	76.0	85
Zinc	31.80	24.0	15
Tin	0.25	....	..
Lead	2.60	0.5	1
	100.45	100.5	101

The hardest compositions are used for the smallest pieces. These are used in the ornamentation of engines, for brass straps, for hinges, and for pulley-sheaves.

Cheap alloys for bearings have been made of the following wide range of composition:

Copper	56	0.5	58
Tin	28	19.5	28
Zinc	16	80.0	14
	100	100.0	100

The first—Fenton's alloy—is said to wear well, not to be specially liable to heating, and to be very durable. The last—Margraff's alloy—is of similar quality. The second composition is much cheaper and lighter, and takes the place of the white alloys used in bearings.

Many cheaper mixtures are used in place of real "Babbitt" metal; and an admixture of lead is often thought an advantage, as it undoubtedly is in bronzes for bearing-surfaces. The softness of metal thus secured permits the journal to take a good bearing throughout, and thus reduced liability to heat as well as to "cut" when actually dry and heating. In common "low-speed engines," with 80 to 100 pounds boiler-pressure and 75 or 80 revolutions per minute, the bearings are usually about two diameters long, and the intensity of their load not far from 150 pounds per square inch of "projected area"; which have about a "factor of safety" against heating, as computed from the formulas of Rankine and the Author, of 3 or 4. In good constructions, the white-metal lining of the bearing extends over the whole surface, is held by grooves in the shell, and is expanded into them by "pening" with the round head of a machinist's hammer.

Guides, bearings, and rubbing parts generally should be brought to size with extreme care, and should be finished very smoothly. It often requires a long period of operation to wear parts not thus at first well finished to a good bearing, and this usually means a very observable waste of power and

loss of efficiency of engine. Rubbing parts of Babbitt or other "anti-friction" alloys do not require as careful fitting and finish; but the companion piece needs all the more perfect finish. It should be made certain, whenever practicable, that the shorter of the two parts, in cases of rectilinear motion, slides completely from end to end of the longer, and even overruns a trifle, to prevent the formation of "shoulders" at the end of its run. Bearings carrying journals heavily loaded should always be made easy fits, and kept quite free at the sides, near the plane of parting, in order that the journal may not be seized, and lateral pressure become so great as to cause excessive friction, wear, abrasion, or "cutting." The final cut, in boring out such bearings, should be made with a broad cutting edge, a coarse fuel, and without stop. The journal should be of a trifle less diameter than the bearing in which it is to turn, and of slightly greater length in order to permit that end-play which is often of advantage in the distribution of the lubricant and insuring uniform and minimum friction.

All bearings under heavily-worked journals, especially at high speeds, should be lined with "white metal," to avoid danger of heating. The general introduction of this class of alloys has very greatly reduced the number of "hot journals" in marine and high-speed engine practice, with corresponding economy of operation, and in repairs, and reduction of personal danger and risks of all kinds.

With great speed of rubbing or with heavy work, eccentric-straps should not be made of bronze, to work on cast-steel. In such cases they should be lined with Babbitt or other white metal.

The cross-head guides are often made of composition, and sometimes are fitted with a water circulation. All parts of the guides should be smoothly and accurately finished, and oil-boxes provided for supplying oil at both ends.

All engine-work not finished should be primed with two coats of brown zinc and oil, and, when placed in position, painted with two coats of any good paint.

Stuffing-boxes made of composition, and fitted with me-

tallic packing, should have efficient means of lubrication. The packing of each stuffing-box is best made in two sections, so that, in case of injury to one, the other may be tight. When large, it is often thought good practice to bush them, at the bearings where the box touches the rod, with white metal.

132. **Rods and Shafts** are being more and more generally made of steel. The larger their dimensions, however, the greater the precaution necessary in selection, in inspection, and in working this material. Care should be taken in the shop, at every step, to see that no defects of material or of structure are allowed to pass.

Seamy iron in shaft-journals is very liable to act as a reamer, cutting the bearing, causing heating, and even sometimes fracture. Rods sliding through packing may exhibit the same defect.

In turning rods, it is important that they shall not be allowed to spring, and thus to assume irregular sections, or a varying diameter. This is especially the case with the piston-rod and with valve-stems. All cylindrical parts should, where practicable, be given good fillets at their junctions with larger portions of rod or shaft, and it should be seen that the tool is not so handled as to gouge in at the fillet, and thus slightly reduce the diameter at that point, there locating a weak section and ultimate fracture. When brought to size by grinding, if emery or corundum is used, care must be taken to see that no grit is left on or imbedded in the finished surface. In steam-engine work, rods should always be given a very easy fit where exposed to steam-heat. The close-fitting of the tool-builder is not desirable. The fitting of the piston-rod in cross-head and piston, on the other hand, demands the greatest care and skill to insure a strong and safe connection and perfect alignment.

Rods and shafts of iron are very often found seamy, and both iron and steel are liable to axial weakness; defects due in the one to defective welding, to crystallization by heat, or to wrenching and defective welding, especially when built up under a light hammer; and in the other to crystallization, or

sometimes to blow-holes and shrinkage. These are often revealed in the processes of finishing, and should always be sought for. The Saxby method of electro-magnetic inspection has sometimes been applied to their detection in important work.*

The seams in the iron always run longitudinally, and do no harm usually, in rods, if the metal be of fairly good quality; in shafts, they not only tend to cut the journal and to produce heating, but the resultant heating exaggerates their action by opening the seams and converting each into an edge, and the shaft into a reamer, to cut down the bearings. All open and perceptible seams should have the edges scraped down so as to do no injury. Thus treated, they may actually aid in reducing friction by distributing the lubricant.† A badly seamed piece should be rejected in all cases. Seamy iron is not always reduced perceptibly in longitudinal strength, but may be weak against torsional stress.

Making rods and shafts exactly to size and perfectly true, both in section and in line, is one of the most difficult problems presented to the builder of the steam-engine; and especially is this the fact when the engine is intended to have high speed of rotation. There is, however, less difficulty experienced in making true the piston-rods and other pieces having longitudinal motion than in securing perfectly true shafts. The main-shaft, the crank-pin, and the wrist-pin at the cross-head must all be made as exactly cylindrical and as perfectly smooth of surface as the art of man can make them. With high-speed engines this is often the great essential to successful operation: the slightest imperfection of surface of journal or bearing, the least departure of the former from exact cylindricity, may entirely destroy all possibility of the satisfactory performance of the engine. For this reason all such journals are now com-

* Materials of Engineering, vol. II; Iron and Steel, § 322, p. 663. Also, Lond. Eng'g, Dec. 1867, p. 550; Mar. 26, 1869, p. 215.

† The Author has, in his own experience on shipboard, seen a shaft-bearing 21 inches in diameter and 2½ feet long thus cut down ¼ inch in a few hours.

monly ground to size in specially designed machines supplied by tool-makers for that purpose.

Such fine workmanship is now considered as essential to a good engine as it is to employ the best of "mild" cast-steel as the material of construction. It is very difficult to secure and to permanently retain perfection of form and finish with wrought-iron, the cinder streaks in which are apt to produce longitudinal lines of roughness. Turning the shafting constantly on its centres, while it is brought to size and given final finish by grinding, is the best method yet devised for obtaining this essential accuracy.

A continuous cut, as illustrated by lathes and milling-machines, is much less costly, measured by the expense of performing work on the tool, than the intermittent cut of the planer. The advantage of the milling-machine over the planer has been found in engine-building, especially in piecework for locomotives, to amount to from ten to twenty per cent. In special cases the milling-machine gives still higher gain.*

In forging iron rods especial care should be taken to "relay the grain" so as to make it follow the outline of the heads as much as possible. This is effected by using more "stock" than would otherwise be employed, and rounding the rod-end to approximately the desired finished shape at a good welding heat. The junction of the head with neck of the rod should be made by a very gradual change of form, and this in the process of forging. Too sharp a reduction of section and the cutting away of the lines of grain by machining is objectionable. "Channelled" or "panelled" rods should be forged with similar care. The panel is, however, milled out, as a matter of economy. The velocity of the cut in planing the rod is usually about 20 feet per minute; the feed, 0.05 to 0.07 inch with common steel; although the use of "self-hardening" steels often permits an increase of work by 50 per cent or more. Bushings are forced into the head of the rod by hydraulic or other presses capable of exerting a pressure of 10 or 15 tons. Keys

* J. J. Grant; Trans. Am. Soc. Mech. Engrs., vol. IX, 1888.

usually taper $\frac{3}{4}$ inch to the foot. Solid ends are becoming more and more usual for locomotive rods.

133. **Valves, Gearing, and Governor** and attachments always demand careful workmanship and, usually, a fine finish. A valve must be made precisely to fit its place; its gearing, where exposed to view, and the governor, must present a pleasing appearance. In making unpacked piston or other balanced valves exactness of size and form and nice fitting demand especial attention. Where packing-rings or adjustable packing-plates are used, precision is less absolutely essential, but is nevertheless important. No careless work is permissible anywhere about this system of mechanism. A close yet easy fit of the valve, movements exact in direction and extent, on the part of the gear, and smooth action of the governor, are all essential to satisfactory performance.

Flat valve-faces are often laboriously scraped to their seats, the latter being as carefully scraped to fit them; but a good machine-tool, leaving a true surface, finished by a wide smooth cut, will give all the accuracy desirable. In either case, the plane of contact being thus established, the friction of the two faces soon substitutes a smooth surface on each, in place of that made in the shop.

Packing-plates, rings, and wedges for balanced valves, demand the nicest work of the engine-builder, and it should be seen that, when finished, they fit well throughout their whole working range.

The more complicated systems of gearing must be not only made carefully to drawing, but it must be made certain that their adjustment and sweep are correct when in place. The governor driving and driven gear must be similarly insured exact and easy action, and the material for its elements must be carefully selected.

In some cases piston-valves are made in a chill, and their working surfaces then ground to size on centres, their seats being often detachable from the cylinder and similarly finished.

Constructing valve-gearing and governor demands, on the whole, perhaps less care and exactness than the making of

crank-pin and fitting of shaft-journals; but it is the most difficult and exacting work on the engine, aside from the latter certainly, and usually gives opportunity to display all the skill and ingenuity of the builder in general fitting. This is usually the most showy part of the machine also, and, properly made, is at once ornamental and striking in effect. All its parts are commonly highly finished, and some brass, and even gold, silver, or nickel, is frequently employed to make a finish and to prevent oxidation. The special precaution to be taken in doing this work is to get every piece of the valve-motion of exact length, in order that the movement of the valve may be just what is intended, and perfectly symmetrical about the transverse centre-line of the valve. Further than this, the valve and its seat should be very nicely made, and scraped to a thoroughly good fit. Once well fitted, if of good design and if well lubricated at all times, no trouble need be anticipated from wear or leakage. Particular care should be taken, where loaded governors are adopted, to see that the inertia of the weight does not introduce liability to wear of parts.

The governor is a piece which is also often supplied by those who make a business of its construction.

In some instances the whole link-motion is case-hardened, and fitted up to exact size by gauge.

Steel pins in "overhung" cranks are to be designed as already described; but usual custom makes them, in the older types of mill-engine, about one half the diameter of shaft, about one fourth that of the cylinder, and their load is often 800 or 1000 pounds on the square inch of projected area of bearing.

For such work at high speeds and great powers as is now constantly illustrated in marine engineering, eccentric straps are usually lined with white metal.

The methods of construction and of setting up a governor are well illustrated in the case of the Ball governor, as seen in the figure. The eccentric *A* is a small disk solidly bolted on the end of the wheel-hub, and as small and light as is consistent with satisfactory operation. The strap *B* is, as usual, in halves; but it is held in place by a thin disk, *C*, on one side

and the not uncommon flange on the other. Links, *DL*, connect the strap and the weights of the governor; while a pin, *E*, set in the cover-plate, actuates the valve as usual. The centre

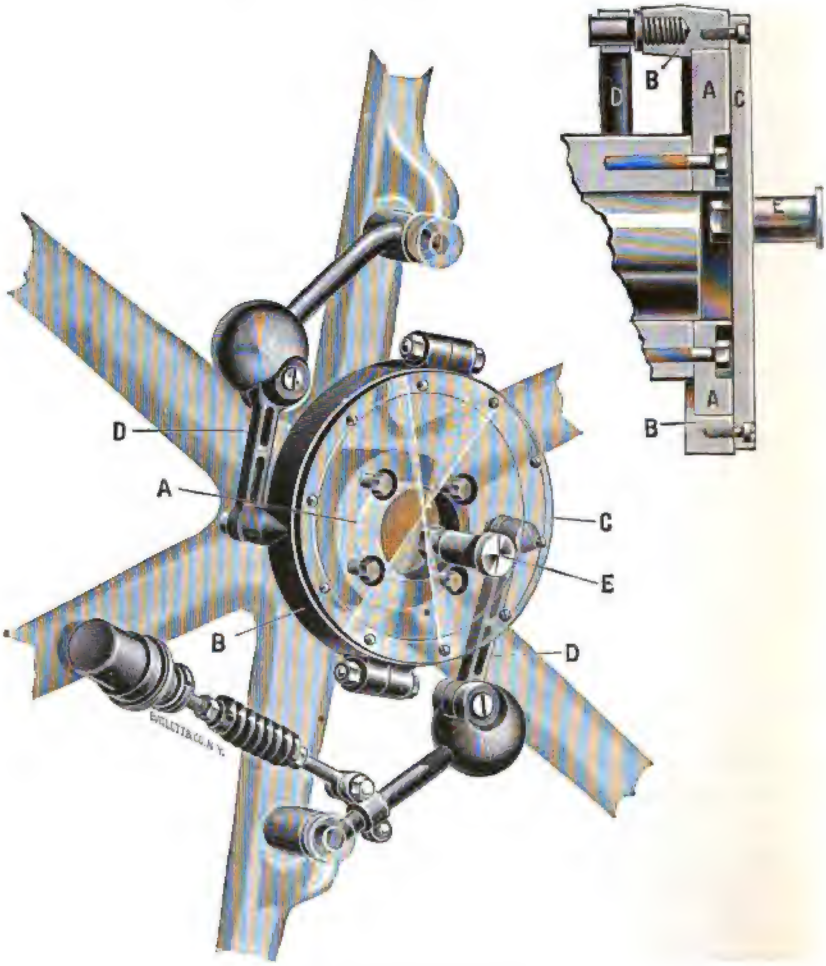

FIG. 186.—BALL'S GOVERNOR.

of the eccentric is so located that its motion relatively to the wheel, as produced by the governor, causes *E* to describe an arc about the eccentric-centre giving the desired variation of

cut-off with nearly constant lead, except at very high ratios of expansion, where the lead is rapidly taken off. When the engine is subject to irregular changes of speed due inertia or gravity in the governor, the dash-pot should be introduced as in the illustration. The method of assemblage is well shown by the peculiar method of engraving the latter figure, a method introduced by Westinghouse.

The diagrams here reproduced illustrate a good adjustment of such a valve-system, and were taken from an engine, designed by Mr. Ball, having four cylinders and of the triple-expansion type, placed in their proper order of relation, and reduced to a common scale. The ranges of temperatures in

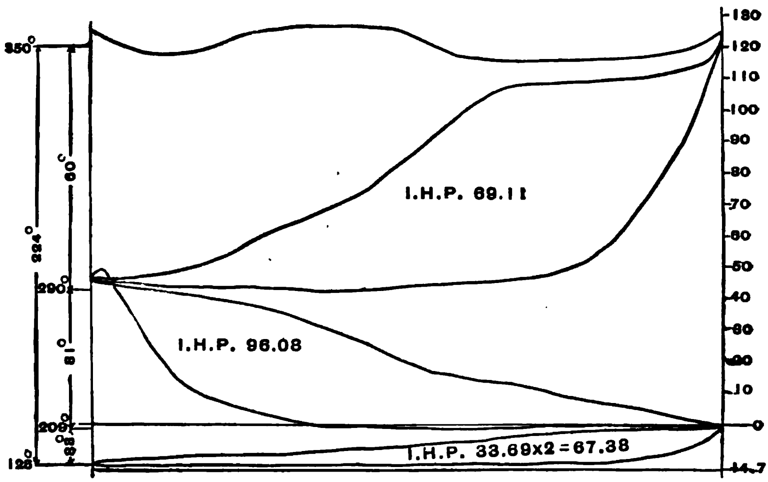

FIG. 187.—TRIPLE-EXPANSION DIAGRAMS.

the several cylinders are also exhibited at the left. In each cylinder the compression is adjusted, as seen, to fill the clearance-spaces, and no appreciable "drop" takes place. The two low-pressure cylinders develop nearly the same amount of power as the highpressure, and the latter about 0.7 as much as the intermediate cylinder. The upper line shown exhibits the steam-chest pressures, and the loss due the throttling action of a long steam-pipe.

134. **Fly-wheel Construction,** in method, depends somewhat on the class of wheel adopted. It is usually a pulley, sometimes simply a balance-wheel, and is rarely, in recent construction, a gear-wheel. When built as a simple balance-wheel, the only important precautions are to secure such exactness of construction as will insure symmetry and good running, and such material and workmanship as will give the full strength demanded by the designer. The wheel should have an exact balance both standing and running, and its rim must run perfectly true. This means the rejection of defective castings, the selection of good iron for bolts, accurate boring of the hub, and, when the rim is finished, a well-turned face. A wheel which does not run true is always a serious disfigurement. The turning of the rim is a matter of necessity in the case of pulley or geared wheels. The former should be carefully turned on face and edges of the rim, and after the several parts, if so built, have been finally and permanently secured together. It should be seen that all bolts are very carefully fitted, that they are well set up, and that no liability exists of their working loose. The rim should be carefully inspected before a beginning is made in turning it, to make sure that adjacent sections are securely united. When turning the face of a pulley-wheel it should be seen that the required degree of "crowning" is secured, in order that the belt shall, on the one hand, not work off, and on the other shall not be distorted by excessive rise along the middle line. This is usually attended to by the designer.

Geared fly-wheels are commonly mortise-gears, and the wheel is usually completely finished before the teeth, or cogs, are touched. The latter are perhaps best of hickory, and are made correct to the drawing by dressing their tips to the right circle, then cutting them to template, one by one, either by hand or by a cutter properly mounted on the bed of the lathe, or at the end of the wheel-pit, in which the gear is finished. A good designer can always readily improvise the needed apparatus in the latter case. The details of this work are given in treatises on machinery and mill-work.

Large wheels are commonly built up of several parts. Those of 12 or 14 feet diameter are often in halves; if of 16 or 18 feet, in six segments; and still larger wheels in eight or even more. Wheels 30 feet or more in diameter have been built, but this means the provision of such a large wheel-pit that few shops even approach this size. For large powers, grooved wheels and rope transmission are often employed. In this case care must be taken to see the proper bevel given the V-shaped grooves on its rim, as given on the drawings.

Fly-wheel construction involves much attention to fitting, usually less to finish. At the high speeds becoming common, the rim, the arms, and the attachment of the latter to rim and hub are liable to be called upon to withstand great stresses due centrifugal forces, as well as inertia-forces brought out by fluctuating speeds. It is for this reason important that the constructor as well as the designer carefully study the methods and details of assemblage. The fitting of the parts of the rim to each other, of arms to rim and hub, and of the hub on the shaft; the setting of the bolts and keys, and all details of the work liable to strain, demand the greatest care and highest skill in construction. Reamed bolts, and keys sufficiently heavy to meet the united effort of centrifugal force and the surges due to varying loads, are required, and should be given a "driving," or at least a snug, fit.

The wheel should be turned, where practicable, on its own shaft, and this should be given a perfect fit. Its rim should run absolutely true; and, if a pulley fly-wheel, the crowning should be made just enough to hold the belt without introducing sensible strains into the latter. If for two or more belts, the successive portions should be separated by grooves deep enough and wide enough to make it certain that well-made belts may work side by side without danger. In such cases the intermediate portions are a trifle higher than where crowned for a single belt. The wheel should be subject to constant and careful inspection while under construction, and when completed, set up and running.

For very high speeds and large sizes gun-iron may be

desirable, and specially constructed joints. When of small size, it should be made in one piece, but very slowly cooled when cast.

135. **Machinery of Transmission,** including gearing or a system of belted counter-shafting and pulleys, is the connecting link between the engine and its work. The power received in violent impulses, usually, is so tempered by the fly-wheel and inertia-effects of the moving parts of the engine, as to reach that system with comparative smoothness of action. It is nevertheless important that the whole shall be so well made, as well as so intelligently designed, that it shall safely sustain such shock as may reach it while under its maximum intended load. This is particularly advisable with gearing, which is liable to injury by violent blows, when with flexible elements, like belting, every shock is tempered and eased until it becomes comparatively harmless.

In no department of construction is it more important to secure good machinery and accurate workmanship than in driving gearing attached to a heavy engine. All the energy wasted by the irregular motion of badly-made gearing is devoted to the wear and the destruction of the gearing; while the jar and noise, if even slightly out of truth, is a serious and sometimes dangerous and ultimately costly matter. Exact templates should always be obtained from the designer and precisely followed in the work. With a good form of tooth, close-fitting, small clearances, and accurate workmanship, fine pitches may be adopted, and a broad face, for large wheels and great power, even, and satisfactory working insured.*

Bevel-gearing is avoided where possible. If made at all for heavy transmissions, it should include a mortise-gear, and all teeth should be planed accurately in special machines like those of Corliss.† All large gearing should include a mortise-gear in each pair. Only the most skilful workmanship and

* The Author once designed gearing of 3½ inches pitch, 24 inches face, an 18-feet mortise-gear, and 9-feet pinion, to transmit 1000 I. H. P., at 50 revolutions per minute, and obtained admirably smooth working.

† See Appleton's Dictionary of Applied Mechanics.

most minute accuracy of form and alignment can secure smoothness of operation in any case.

The loss of power by gearing and by belting is variable with the proportions and arrangement of the gears and pulleys, length of belt, etc. It averages not far from 20 per cent for a single pair of bevel gears, uncut and dry, though smooth for such gearing, and but 10 per cent for the same well lubricated.

Belting and rope-transmission exact far less nicety of construction; but their pulleys and shafts must be carefully inspected as to material, workmanship, and dimensions to give the system smoothness and durability of operation, and to give true-running pulleys and belt. Care is especially demanded where great power is transmitted through belts either in multiple or in superposed series, and when, several ropes being required, all must pull together and without vibration.

136. Machinery of Condensation, including condenser, air-pump, and minor accessories, is often supplied independently of the engine itself, and constitutes a system the character and proportions of which are determined by the designer after a study of the locality in which it is to be used, and the facilities for supply and discharge of the condensing water. The builder is only concerned with the construction of the apparatus in accordance with the drawings and specifications, and will apply to the task precisely those principles already enunciated in that connection.

The condenser must be a sound casting, the joints well made, and the nozzles and flanges between it and the air-pump secure. The pump-barrel must be made with the same precaution as the steam-cylinder, but not necessarily of as hard iron as is commonly demanded for the latter. Its interior is often lined with brass or bronze, which takes the wear. This is nicely fitted in parts, like the staves of a barrel, if large, or a cylinder nicely turned both inside and out, and forced snugly into place. The former is always, the latter often, "pened" by blows of a light hammer, until it is well expanded against the enclosing barrel, and is then bored to size. The valves are sometimes of metal, sometimes of rubber or of canvas,

and must open and close easily, and be tight when shut. A metal valve faced with rubber is often found a good form. The valve-seats and the bearing-faces of the valves, if of metal, should be perfectly smooth and sound; otherwise they are liable to leak after a short period of use, and to make serious trouble later. Good bronze gun-metal may be used for valves. If liquation has taken place, indicated by variations of surface color and scattering tin spots, the casting should be condemned.

Should the system not work satisfactorily when set up, if any large rubber valves are used, it may be found that they have become distorted by the heat of the water passing through the pump. Small valves give less trouble in this respect than large; and their substitution in greater number for the larger size may permit the operation of the condenser at higher temperature than might be otherwise practicable,—a desirable matter, often, where high steam-pressure is adopted.

Machinery of condensation under construction demands, as a rule, less attention to secure perfect fits and smoothness of rubbing surfaces than other parts of the machine; as the air-pump and the circulating or cold-water pumps, if the latter are used, are very usually driven under lighter loads, and, wherever practicable, at lower velocities of relative motion than the main engine. These elements should, nevertheless, be carefully made, well fitted and well finished; while special care should be taken to see that every precaution is observed, in construction and in assembling, to insure against liability of leakage in the pumps or in the condenser, where the surface-condenser is employed, as is advisable when the feed-water is expected to be seriously impure. The fitting of the valves and seats in the pumps, and the packing of condenser-tubes in the latter case, are the main objects of precaution in building this class of machinery.

137. **The Use of Drawings** by the builder of the steam-engine is perhaps more essential to correct and economical construction than in almost any other mechanical work. The designer should furnish drawings, complete, both general and detail, and should be held responsible for their accuracy. A

complete set of drawings includes elevations and plans of engine and of foundations; drawings of valve-gearing showing its construction and its adjustments; and sheets exhibiting all details, and the finished sizes. Drawings are commonly made to scale, and sizes are also figured on dimension lines. In all cases, it is customary to instruct the workman to work to the figures, and never measure the drawing where figures are given, as alterations of dimensions are often made after the drawings are completed. The scale selected is, for large parts and general drawings, such as will enable the designer to put his plan on a convenient size of sheet, and for small parts, such as will permit easy reading of the drawing and taking off of sizes by the workman. It is customary, in the best-arranged shops, to use a few standard sizes of sheets, usually having common multiple dimensions, and to have all drawings made to those sizes.

Original drawings should always be retained by the designer, copies being made, by tracing or by blue-printing, for the builder and the shops.* The originals are often only made in pencil, and their reproduction by the now common photo processes is relied upon to give permanent records. When, in making a set of drawings, any doubt is felt as to the clearness of the representation of any part, a sufficient number of views should be made to insure that the builder and his workmen shall fully comprehend its every detail; and it should also be shown as connected with the adjacent members of the machine, if such a sketch can be made, to render it more certain that its form, size, and use will be more perfectly understood.

Drawings supplied by the designer are expected to be full and complete in all respects; but it is often found that matters of detail are left to be worked out by the constructor, and it is also sometimes discovered that errors exist, and even that the drawings are self-contradictory. In all cases the responsi-

* With good paper and tracing cloth and *pure* reagents, equal parts of ammonio-citrate of iron and ferrocyanide of potassium in four times their united weight of water, and an exposure, in a clear day, of eight to ten minutes, will make excellent prints.

bility lies with those who furnish the drawings, and, in matters of doubt, they are to be exactly followed on the assumption that the reasons for apparently singular or false construction will appear in due time; but the builder will always watch for errors, and, detecting anything that seems to be wrong, will instantly seek instructions, if it is possible to reach the designer. Where anything is unquestionably and obviously wrong, he will leave the correction to the designer if practicable. If compelled to act on his own judgment. he will do so with the utmost caution.

Where details are left to be put in by the builder, he will use his own discretion, but will ordinarily adopt well-known and successful devices, and will attempt nothing experimental or novel, unless with a clear understanding and full approval of the designer or proprietor. In important and costly constructions, it is wise to secure a written order, or even a new contract with drawing and specification, before venturing upon the work of construction of parts not fully described and drawn in the original specifications.

When the drawings are first received, they should be at once carefully studied with a view to ascertaining if they accord with the contract-specifications, are complete, intelligible, accurate, and convenient in form, and the distribution of detail on the various sheets. If found in any way defective, they should be immediately returned for amendment before beginning work with them.

In some cases the builder is allowed to introduce some special, often a patented, device of his own invention, or of which he is proprietor. Thus, Messrs. Bates & Co. of Sowerby Bridge (G. B.) oil the main bearings of all their larger triple-expansion engines by fitting each with a small pump set in an oil-tank receiving the returning oil.* This is an example of special practice. Messrs. Cramp & Sons introduce into their marine engines a patented form of reversing gear and other details, illustrating the second case.

* D. K. Clark; Steam-engine, p. 463, vol. II.

It should be seen that the several parts will fit each other, that the forms and dimensions are correct and usual, that the sizes are everywhere marked, that finished parts are properly indicated, and that a sufficient number of views of each piece is given, so that no uncertainty can anywhere appear. It is always better that, after a first examination by each, a revision of the drawings and specifications be made by the designer and the builder together, and all uncertainties removed, should any arise, either as to what is meant by them, or whether the construction is practicable, satisfactory, and reasonably inexpensive.

The drawings supplied are usually on a reduced scale. The builder often desires some or all parts reproduced full size. In such instances he commonly makes his own enlargements, and is himself responsible for their accuracy.

The builder's tools require careful attention, if he is to do work economically, secure good contracts, and fulfil his agreements with advantage both to his customer and to himself. The choice of material, of speeds and depth of cut, and the form of tool and method of tempering it, all demand careful consideration to insure maximum economy and good work.

In shaping and grinding cutting tools experience indicates the following as good forms; the "rake" being the angle between the normal to the surface cut, at the point of the tool and the line of the cutting and parting face; the "cutting-angle" being that between the latter and the tangent to the surface at the point of the tool; and the "bottom," or "clearance angle," that between that tangent and the lower and outer face of the tool:

TOTAL ANGLES FOR ROUGH WORK.

Material cut.	Rake.	Cutting-angle.	Clearance.
Cast-iron, soft	15°	70°	5 to 10°
Cast-iron, hard	15°	75°	5°
Wrought-iron	35°	50°	5°
Mild steel	30°	55°	5°
Hard steel	25°	60°	5°
Brass and bronze, soft	5°	80°	5°
Brass and bronze, hard	5°	85°	0°

The tool is set to cut above the plane through the axis in turning wrought-iron and steel, at that level for cast-iron, and a little under it with brass work.

When hardening steel tools a very uniform heat is required for pieces of uniform section, and higher heat is necessary as the size of its section increases; but this difference should not be great, and it should be very carefully graded, as a high heat produces a coarse, open grain, and irregularity of heating is likely to cause cracking from internal strain. Cooling should be moderately rapid, complete, and perfectly regular. The bath should be large, and supplied with running water when large pieces are to be hardened. Tempering should be done very carefully, and the cooling should take place slowly and very regularly throughout the piece. The lowest heat at which the steel will harden is best. Hot steel, if not intended to be tempered, should always be cooled slowly, and in a dry place of uniform temperature. When annealing, heating slowly to a low heat, and cooling in ashes or other non-conducting material, gives the best results.*

Pieces of very small section are sometimes tempered by contact with a smooth surface of cold metal; they may be annealed by cooling slowly in contact with, and simultaneously with, a larger mass of heated iron. Charcoal is the best fuel for use in working steel, and coke is better than coal.

To prevent loss of carbon, small articles are often covered with a flux having carbonizing or deoxidizing properties. Watch-springs are sometimes heated in molten glass, and larger articles in a bath of melted metal, as lead.

In drawing the temper, the hardened steel is usually reheated until the scale of oxide on the surface assumes a certain color; which color indicates a certain temperature which is constant, or nearly so, for any one steel, and is slightly different for different steels. The colors and the tempers so obtained are usually given as below:

* Materials of Engineering; Iron and Steel; vol. II, §§ 187–190, pp. 317–328.

Temperature.		Color.	Use.
Cent.	Fahr.		
220°	428°	Pale yellow.	Surgical instruments.
230°	446°	Straw.	Penknives, razors; wood-tools.
255°	491°	Brown yellow.	Chisels and scissors.
265°	509°	Purplish.	Axes; heavy knives.
275°	527°	Purple.	Table-knives; springs.
290°	534°	Pale blue.	Watch-springs; swords.
295°	563°	Dark blue.	Fine saws; drills.
315°	600°	Very dark blue.	Hand-saws.
350°	662°	Very dark blue, verging on green.	Too soft for any ordinary tools.

A good oil composition for hardening consists of—

Spermaceti oil	48	parts.
Neats'-foot oil	47	"
Rendered beef suet	4	"
Resin	1	"

The tank in which it is placed should have a close-fitting cover, which will put out the blaze when the tempering is finished.

The colors adopted are not invariable for even the same purposes. Watchmakers' tools are heated in the flame of a blow-pipe or of a lamp, and are hardened either in the air, or by plunging their points into wax or tallow. Saws and springs are often hardened in mixtures of wax and oil, tallow or suet; the tempering is done by "blazing off" the grease. Car-springs and carriage-springs are heated to a low red heat, cooled in hot or in warm oil, and left without further tempering. Large pieces must be "drawn" more in tempering than small ones.

Chrome steel may be forged like any other; but all tools drawn from a large body to an edge should be allowed to cool off after forging, and should be reheated for tempering, as the interior of the mass retains the heat at which it was forged long after the external surface has cooled. It is still too hot for tempering, and is liable to crack on cooling.

The following table shows more generally the proper treat-

ment of tool-steels in tempering for various purposes, water being used in cooling:*

SCALE FOR TEMPERING TOOLS OF CARBON STEEL.

Material Cut.	Tools urged by									Nomenclature.
	Pressure.				Impact.					
	A	B	C	D	E	F	G	H	K	
Unannealed steel.	0	1	2	2	3	4	2	2	7	0. To remain as dipped.
Annealed steel.	1	2	2	3	3	5	3			1. Light-straw color. 2. Dark-straw color.
Chilled cast-iron.	0	0								3. Orange color. 4. Reddish-purple color.
Hard cast-iron.	0	2	3	3	4	2				5. Purple color. 6. Bluish-purple color.
Soft cast-iron.	1	3	3	3	5	3				7. Dark-blue color. 8. Light-blue color.
Gun-metal (bronze).	1	2	3	3	6	6				9. Bluish-gray color. 10. Soft.
Yellow brass.	2	3	3	3	6	6				A. Turning or planing tools.
Soft composition.	3	4	3	3	7	7				B. Drills, bits. C. Taps, dies.
Wrought-iron.	3	6	3	3	7	7	4			D. Rimmers. E. Cold chisels.
Copper.	4	6	3	3	7	7	4			F. Flogging chisels. G. Calking tools.
Wood.	6	6								H. Hammers. K. Springs.

Chrome steels, and also the wolfram steels, should be hardened at the lowest heat possible.

Finishing iron is performed with a broad-nosed tool and rapid "feed." For brass-work, a smaller, round-nosed tool is

* Ibid.

employed. The form of the body or shank of the tool is often determined by the shape of the work.

The speeds of cutting are usually not far from the following, in feet per minute, as accepted maxima:

Material.	Boring and Turning.	Planing.	Milling.
Cast-iron	15	15	50
Wrought-iron	20	20	60
Mild steel	15	18	60
Tool "	15	15	45
Brass-work	50	40	80

By selecting soft and uniform metals and alloys these speeds may be increased, and by the employment of the "self-hardening" tungsten, chrome, and other "steels" very greatly exceeded. The depth of cut is determined by the strength of the piece and the machine in which it is cut. It varies from $\frac{1}{16}$th or $\frac{1}{8}$th inch on ordinary work up to $\frac{1}{2}$ inch and even over an inch on heavy work, as in taking a roughing cut in boring or facing off large engine-cylinders, or in turning heavy ordnance.

Tools of cast-iron may often be substituted with advantage for those of steel. When exceptionally hard iron must be cut, as in boring hard steam-engine cylinders, chilled iron is very frequently employed as a last resort, and tools of the usual forms are sometimes used in lathes and planers, having somewhat larger size of shank than those made of forged steel, and with their cutting edges chilled. The cut may be heavier, and the speed greater, than with steel tools.* They are not suitable for finishing, being liable to lose their fineness and smoothness of edge, but are preferred for roughing cuts. Dr. Dudley finds the mixture used in the shops of the Pennsylvania Co. at Altoona to be mixed as below, and to have the following composition, both for wheels and tools:

* Cast-iron Tools, by Oberlin Smith; Trans. Am. Inst. Min. Engrs., 1890.

CAST-IRON CHILLED CAR-WHEELS AND CUTTING-TOOLS.

Brand.	Kind.	Per cent used of each Brand.	Chemical Analysis.				
			C.	Mn.	P.	Si.	S.
Greenwood........	Charcoal	5	3.61	0.41	0.43	0.74	....
Lime Rock.........	"	10	3.60	0.85	0.33	0.98	....
Richmond........	"	5	3.52	1.09	0.28	0.92	....
Shelby............	"	5	3.61	0.23	0.46	0.70	....
Raven's Cliff.......	"	5	3.89	0.27	0.16	0.50	....
Glendon...........	Anthracite	10	3.65	0.41	0.54	1.23	0.03
Steel rail..........	"	5	0.40	0.85	0.09	0.03	0.03
Old wheels.........	"	55	3.50	0.52	0.42	0.75	0.08

138. The Assemblage of Parts, if the machine is of large size, is ordinarily carried on in a special erecting-shop. Very large engines sometimes occupy too much space to be fully erected until they are permanently in place for work. In the former case, opportunity is allowed for the exact fitting and adjustment of part to part at little cost and most conveniently. All errors of design and construction then usually become apparent, and the machine is only sent out after they have been corrected. In the latter case, it is especially important that no such errors are allowed to pass undetected in the shop, as their correction later may be a matter of great cost, and very serious delay and annoyance.

In fitting up the fixed parts and in adjusting the moving parts the essential precautions are to see that the former are supported on a sufficiently solid base and in correct relation and alignment, and that the latter are fitted together well and neatly, and in such manner that they may traverse their cycles of relative motion accurately, easily, and smoothly. This process also gives an opportunity for a final inspection of all parts, and of the workmanship and finish, as well as of the material. It often happens that defects of material and construction are only detected after the work is done, and errors in design come to light when the machine is assembled.

Gauge-work is usually seen where many pieces are made alike, and, once the gauges are tested and found right, the parts may be expected always to go together right. Special ma-

chinery and interchangeable parts give enormous economical advantage where sufficient numbers of the pieces made can be disposed of.

Where a single machine is built, the exact size of any one piece, within the limits of ordinary variations occurring in the shop, is a small matter generally, as one piece may be fitted to that with which it is at any point paired by a system of adjustment of the one to the other, which may often be made easily to effect a compensation for any observable variation of either from the correct and prescribed size. Where considerable numbers are to be made from the same designs, on the other hand, this method of cutting and trying is not permissible, and every piece must be made exactly to size and to gauge, in such manner as to make all similar parts interchangeable, and equally well fitted to every other opposite element of each pair. This modern system of "making to gauge" and of interchangeability, where many similar machines are to be built, is found very greatly to facilitate rapid work. In the case of machinery liable to occasional accident, as is the steam-engine very generally, this also permits the builder to supply duplicate parts at once and at small expense to either vendor or buyer, and thus often to enable the latter to evade those enormous expenses coming of interrupted production when the prime mover is disabled. This system has, for these reasons, come very extensively into vogue among constructors of steam-engines.

Where this is done, parts are made in lots of greater or less number, according to the extent of the business; and a few engines, often but one or two, are set up on the floor together. Often the smaller and the more intricate parts or combinations of elements are kept in stock, or sometimes the cylinder and valye-gear; while the other and heavier parts are only made as orders come in.

139. **Erection of Engines** in place, and their preliminary operation, is finally intrusted to the best and most experienced workmen. It is a task to be performed, often under great disadvantages, far from the builder, and even, it may be, remote from any workshops. It is a simple process, but one

demanding judgment, experience, familiarity with the construction of the machine and with its purpose and its action. Fertility of expedients, coolness in emergencies, and readiness in accommodating himself to circumstances are essential qualities, also, of the person in charge of this work.

Foundations are to be built in accordance with the drawings furnished, and, if necessary, with such modification as may be exacted by unforeseen peculiarities of location.

It is best never to use lime mortar in place of hydraulic cement. In preparing for foundations the essential requirements are solidity and permanence.

On swampy land or quicksand drive piling until stability is assured. Where piling cannot be used, lay down heavy plank 3 to 6 inches in total thickness, preferably double, and build up with concrete to the surface, and finish with brick or stone laid in cement. For dry clay or gravel no such precautions are needed. Hard burned brick, laid in cement, is a very good foundation. It is unsafe to build a foundation on frozen ground or where the cement will freeze before setting.

If it is already built, it must be inspected to see that it is on a safe bed, measured up to make sure that it is of correct proportions and dimensions, and carefully examined to ascertain if it is constructed of the prescribed materials and in a workmanlike manner. These points satisfactorily settled, the engine bed or frame is put in place, levelled up, foundation-bolts made secure, and tested, finally, to insure correct position on its foundation and exact alignment with the machinery of transmission. So much being accomplished, the various parts of the machine simply fall into the places assigned them, and as fitted in the shop, and the completion of the work of erection goes rapidly forward.

Making connections of engine with boilers, and, if condensing, with the water-supply and delivery pipes, may sometimes demand the construction of a complete plan of piping; but this should have been already made by the designer, and pipe supplied in accordance with his plans. The great art in erection is usually that of getting the centre lines of the engine

and its system of condensation accurately established; this done, and the shop-work being accurate, the rest is easy.

When finally completely assembled, the engine is cautiously "turned over" by hand, to see all clear and in working order, and that every part moves as intended, and easily, yet without "lost motion." Steam is then slowly let on, and the engine moved under steam and with a minimum load. If all goes right, the load is gradually brought up to the maximum, and the machine is transferred to its owner, as soon as the builder has satisfied him and himself that the contract is fully carried into effect.

Every type, size, and class of engine has its own special and appropriate method of treatment; but the general method and the main operations are the same for all. The locomotive is erected on the track, its boiler and frame serving for foundation and bed-plate; the marine engine is carried by the frames of the ship and the main and special engine keelsons; and these and the machinery must be designed and constructed to fit each other. The plumb-line may be used on stationary engines in their erection; but it cannot be used on shipboard, and the rule-square must be substituted. For each case, experience, judgment, and ingenuity come in play.

In erecting marine engines the following is a very usual system:*

The keelsons supporting the propeller-shaft are parallel with the load-line and the fore-and-aft line of the vessel.

The constructing engineer takes this as the reference-line longitudinally, and the deck-beams as his guide transversely. The main-shaft line is that from which all others are obtained.

The constructing engineer gives the plans of the cross-keelsons, etc., which serve as the foundation of the engines. When the hull work permits, the engineer lays down the bed-plate by a template. The centre-line of shaft is marked off on board ship, and the bolt-holes marked off to receive the bed-plates.

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*See The Cadet Engineer; Long & Buel; J. B. Lippincott & Co., Philadelphia; 1866.

To find the fore-and-aft line, get the centre from outside to outside of planking, and draw it fore and aft on the deck. Draw the athwartship line at right angles to this line, and parallel with the deck. Next transfer the fore-and-aft line to the main keelson, and adjust the engine-keelsons parallel with it, and level off the keelsons. The centre of the condenser must be made to coincide with the fore-and-aft line. The gallows frame of a beam-engine is adjusted to a line drawn from the lower fore-and-aft line, at right angles with the shaft-line.

The boiler-keelsons are made parallel with the load-line and the deck-beams. Oak should not be placed in contact with the iron, as its acid will destroy the iron. Yellow-pine is preferable.

As the surfaces of boilers are never smooth, putty is generally used to fill up irregular places. They are firmly bolted to the hull.

Holes bored through wooden sides for pipes and cocks should be lined with lead and bushed with copper. The hole for the propeller-shaft should be accurately bored with a boring bar, and then lined with lead forced by dies close against the wood, and then lined with copper similarly fitted.

Small engines are usually completed, fully erected, in the shop, and given a trial with brake and indicator to see that they are capable of doing full work without undue friction, and with the desired accuracy of regulation under varying load and steam-pressure. They are often, on these occasions, kept in operation hours, or even, if their journals heat or their governors work badly, for days, and until all defects so revealed are rectified, and until the engine can be well adjusted as to balance and to steam-distribution.

Formal engine-trials are often made by or for the purchaser to determine whether the guarantees of the builder are fully met. These are often somewhat extended scientific investigations, and of great interest and importance.

CHAPTER V.

OPERATION, CARE, AND MANAGEMENT.

140. The Operation and Care of Engines and Boilers are confided to a class of men of usually entirely different character and attainments from those concerned in their construction. Theirs is an art distinctively different from any other. It is usual and indeed important, if not essential, that the engine-driver, or the engineer, as he is often called, should be perfectly familiar with the structure of his engine and principles of its construction, and that he should also be a good mechanic and competent to make his own repairs; he must be ingenious in devising means of reconstruction under the most disadvantageous conditions and the most discouraging circumstances. His ordinary vocation, however, except on the locomotive, is simply the operation of the machinery and its care under the everyday conditions of its use. Even this, however, exacts care and never-ceasing vigilance, and absolute integrity and trustworthiness are his most essential characteristics. The care of a locomotive demands, more than perhaps any other situation known in civil life, a combination of intelligence, courage, self-confidence, quickness of mind, eye, and hand, readiness in emergency, and absolute steadiness, such as should command the respect and admiration of all men.

So far as the care of the machinery is concerned, the demand is the same in every case: as to the boiler, a safe and steady water-level in all, even firing, constant steam-pressure; as to the engine, good lubrication, and careful provision against water entering the steam-cylinder, in consequence of carelessness either at the boiler or at the engine.

Stationary engines of less than about two hundred horse-power, as a rule, are placed in charge of a single "engineer," who is also expected to manage the boiler and to do the "firing" or "stoking" without assistance. Marine and other engines which are necessarily in operation night and day are operated by a corps of engineers and firemen working in "watches," either in alternate "gangs" or "crews," or "four hours on and eight hours off." With large engines, the engineer's department is manned by a crew consisting of a chief engineer, several assistant engineers, water-tenders and "firemen" or "stokers," and "coal-passers" or "coal-heavers." In such cases the number of men on watch is determined by the character of the machinery and its complexity, by the quantity of steam to be made and coal to be handled, and by the temperature and more or less perfect ventilation of the "fire-room" or "stoke-hole." The number of firemen should ordinarily be such that none need be required to handle more than about a ton of coal per hour; and the number of coal-heavers should not be less than one to every two firemen. Water-tenders are assigned, on large and hard-driven steamers, one to every six to ten furnaces. One engineer is assigned to each watch, and is assisted by one "oiler," and sometimes by two, or by an apprentice or assistant if the engines are very large, or if auxiliary and independent engines are used. On large naval vessels, where numerous anchor-hoisting, blowing, turret, dynamo, and reversing engines are employed, considerable accessions to the usual force may be demanded. A chief engineer has general charge, and is held responsible for the engineers' crew and for the efficient working of the whole department.

141. **The Responsibilities of the Engineers** and their assistants are rarely very precisely defined, except in the naval service. Every vessel-of-war of any considerable power and size is such a mass of machinery, taken as a whole is such a machine in itself as well as in its elements, that it must be supplied with a large body, not of engine-drivers simply, but of the members of an educated engineer corps, familiar with the

principles as well as the practice of their vocation. In the apportionment of their duties minute regulations and instructions have been introduced and carefully adapted to the various conditions existing on the several classes of ships. The following is an example of a concise and simple statement of the main duties and responsibilities of these officers on board a ship of small size. This was in part the code governing the engineer department of a small U. S. naval steamer, of which the Author was engineer-in-charge during the civil war (1861–1865):

Instructions.—The Chief Engineer will have charge of, and be personally responsible for, the machinery and all property of the engineer department; he will assign the subordinate officers and the crew to their duties and positions, and will institute such regulations as may, in his opinion, be required to insure proper discipline and the highest efficiency of the department. He will report to the Commanding Officer, and by the latter will be held responsible for the proper operation and preservation of the machinery of the ship, and for the preservation of good order in his department.

The Senior Assistant Engineer will have charge of the engines, and will attend personally to the making of repairs, packing, and adjusting, and will in all cases keep the Chief Engineer advised of the condition of his work, and the method and progress of repairs.

The engineer next in rank will have charge of the boilers, propeller and shafting, shaft-alley, fire-tools, and all boiler and miscellaneous apparatus not otherwise assigned. The repairs of the boilers will be made under his personal supervision, and with the advice of, and in consultation with, the Chief Engineer.

The next junior engineer will have immediate charge of the blowing-engines, the coal-bunkers, the coaling apparatus, of the logs, the stores, tools, and outfit, and will be expected to take an inventory of all property of the department as often as required by the senior engineer.

When not under steam, the assistant-engineers will keep

regular days-duty, commencing at 8 A.M. of one day, and ending at 8 A.M. of the next, each in his turn, as assigned by the senior. The engineer on duty will see that the men are at their work promptly and during the whole period of the working day, and that their work is well performed, that the stores are not wasted, and that the tools are properly used and cared for. He will go on duty when all hands are called, and will see that the men are all at their posts within twenty minutes thereafter. He will assign reliefs for such as are known to be sick, and will report absentees to the senior engineer. He will not permit the men to leave their stations before "seven bells," and will see that the department is ready for inspection when "two bells" are struck. . . . He will be careful to note on the log-slate the nature and progress of all work going on under his direction, and will, in addition, record every event of importance occurring during his period of duty. At least one engineer must be on board ship at all times, ready for duty.

When orders are given to raise steam, the day's duty ends. The senior assistant engineer will take the first watch, starting fires, opening the log regularly, accounting for all coal and stores employed, seeing that the oiler or the water-tender reports the height of the water in the boilers, at starting the engines and at stopping, and observing anything that may need attention about them. The engineer of the watch will be careful to keep every column of the log-book entered up, and will be responsible for the operation of engines and boilers during his watch. He will himself examine the saturation of the water in the boilers, and the height of water in the gauges, and will in no case allow himself to be relieved without first seeing that the water in the boilers is at the right point and of proper density. The relief will also note the same points, and will at once report to the Chief Engineer should he find anything wrong, either about boilers or engines. No officer will be allowed to leave his station until regularly relieved; and no fireman or coal passer shall leave without the permission of the engineer on watch.

All the men, on coming on watch, will report themselves

to the engineer of the watch, will similarly report when relieved, and will inform their reliefs where they may be found if needed during the night. The engineer will not permit the men to handle the valves of boilers or engines; but he will himself see them properly adjusted. All reports will be made to the Chief Engineer, who will expect regular reports from each of the junior officers having charge of sub-departments each evening at eight o'clock. Assistants desiring to go on shore or to obtain leave of absence will first secure the approval of the Chief Engineer, and will report to him at once on returning to the ship.

When in action, the senior assistant will take the watch and handle the engines, the next in rank will take charge of the fireroom, and the junior will go to the bell. If either be disabled, the next in rank will take his place, and in case of disability of so many of the staff that each station cannot be filled, the best among the warrant officers, or among the enlisted men, will be assigned to the duty by the senior engineer on duty. One assistant and the storekeeper will, when practicable, be on duty in the storeroom, ready to serve out such tools or stores as may be suddenly called for in case of accident to any part of the machinery. Those disabled will be at once taken to the surgeon, if possible, the engineer-in-charge detailing men to that duty, and reporting the facts at once to the senior engineer.

The "engineer" in charge of the locomotive has less complicated machinery to handle; but his responsibilities are even greater and his duties much more exacting than those of the marine-engine driver. In either position the qualities demanded are of the highest character. He must have no inconsiderable intellectual ability, experience, moral strength, and physical power of endurance. He must be familiar with the construction of the engine and all ordinary methods of preservation and repair; he must understand the essential principles of its philosophy and operation, and the reasons for every rule of practice which he is expected to observe. The locomotive-engine driver must be able to watch his engine and the track

at the same time; keeping an eye on the steam-gauge, trying the gauge-cocks, handling the pumps, the throttle, and the reversing lever, directing the fireman, and keeping note of the speed of the train and its location; all the time alert, cool, self-poised, and ready. He should have often thought over the wisest ways of meeting emergencies, and will often, in such case, act intuitively the instant the occasion arises. A cool head, quick perceptions, and sharp senses will ordinarily reduce the risks to a minimum, if coupled with decision, energy, and a good judgment. Only such men should assume or be admitted to such positions. The engine-driver should be a good mechanic, familiar with shop-ways and methods, a skilful fireman, neat in his person and work, and prompt in meeting his appointments. The locomotive-driver must also be competent to show his fireman how to make up the fire, and to handle it when on the road, should know the road and the signals perfectly, and that exact manipulation suiting the action of the engine to the demands of the train, as affected by speed, weather, gradients, and weights hauled.

The management of the locomotive thus demands more care, attention, and judgment, and more presence of mind in emergencies, than even that of the marine engine.* On taking charge, the water in the boiler is looked to, the supply of sand, of oil, and that of fuel and water in the tender, examined; the machinery and lubricating apparatus is carefully inspected, and the condition of the engine and boiler generally seen to be satisfactory, including the tools and spare parts, and the lights and signals.

In case of accident on the road, the first step is to see that approaching trains on either side are warned promptly, and, if possible, notification sent to headquarters. On heavy gradients the water must be carried at such a height that no danger of uncovering any heating-surface can arise; ordinarily it is carried "steam and water at the upper gauge-cock." If foam-

* For details of management, see "Progressive Examinations," by John A. Hill, N.Y., 1891; and especially Forney's "Catechism."

ing, consequent upon the presence of oily or mucilaginous or other foreign matter, occurs, or if priming takes place from hard driving or deficient steam-space, it must be carried lower. If the level falls dangerously low from any cause, the fire must be drawn at once.

When one side of the engine is disabled, a common occurrence, it is disconnected, and the other side will sometimes do the work until the engine can be relieved. Of the numberless possible minor accidents, each demands its own special remedy, and forethought usually provides for its prompt application.

Where continuous brakes are fitted, they are inspected before starting and at the end of the run.* They should always, unless in a serious emergency, be applied as gently as possible, and removed immediately when no longer needed.

142. **Engineer's Inspections** may be special, or regular and prescribed as a part of current duty. On taking charge of the machinery on his first appointment, the engineer-in-charge makes a complete and minute inspection of every portion of his department and of every detail of engines and boilers. If any defects are discovered, he should instantly report them to the person to whom he is held responsible. This inspection, and subsequent repeated examinations and study of the machinery, enable him to become thoroughly and promptly familiar with the construction, the condition, and the method of operation of every part, and to determine what, if any, alterations or repairs may be needed to increase its efficiency or to make it safe against accidents.

Such inspection should also be made at intervals so short as to make it practically certain that dangerous deterioration shall not occur, unperceived, in the interval, and repairs discovered to be advisable or necessary should be made at the earliest possible moment. Reports on the condition and needs, immediate and remote, of every detail should be made to the proprietor or manager, or, if at sea, to the immediately

* See "Instruction Book" of the Westinghouse Air-brake Co.; Pittsburg, 1890.

superior officer, at regular intervals. Special reports are required whenever an emergency may make it advisable. These reports should include all data and available information relative to performance and the economical operation of the machinery. Accidents should be instantly reported, with a statement in detail of cause and remedy.

After every period of prolonged continuous operation, all parts of the machinery should be completely overhauled and dismantled sufficiently to permit complete inspection of every piece and of every journal and bearing, however much time and trouble may be involved. Intermittently working machinery should be similarly examined at intervals, taking advantage of opportunities offered when regularly off duty.

The special purpose of the inspection should be to see that every part of the engine is in safe condition and ready for immediate and satisfactory use; that all connecting parts, as bolts and nuts, pins, ends of rods, journals and their bearings, are in good order and of ample strength; that joints are tight, stuffing-boxes well packed, and valves and seats smooth and "tight," and all easy working. The boilers should be examined and tested to see that they do not leak, and to make certain that they are capable of sustaining at least twice the working pressure. The safety-valves should be particularly examined; and it should be seen that they are of sufficient size, of good form, and working freely to the full limit of their lift. All the machinery and piping connected with the feed-system should be seen to be safe, convenient in arrangement, and amply powerful for any emergency.

If the machinery is in operation when the engineer-in-charge enters upon his duties, he can do little more than see that the water is at the right level in the boilers, the feed-pumps working properly, and all the engine-journals working cool. He should make sure that the air-pump—if it be a condensing-engine—is working well, the condenser and hot-well at a good temperature, and the vacuum such as should correspond to it. He may also take advantage of the opportunity to judge the capacity, efficiency, and temperament of his men, and settle in

his own mind their fitness for the duties to which they may be severally assigned.

In the naval service,—and properly wherever there are several such officers,—as prescribed by regulation, "the senior engineer officer on board, in the absence of the Chief Engineer, or in ships where no chief engineer is borne, is to observe and follow the instructions for that officer, and to consider himself responsible for the care and proper working of the engines and all parts connected with them."

The engineer-in-charge should make a careful and complete inspection for his own information and satisfaction before accepting any responsibility for the condition of the engines, boilers, or accessories. When the machinery is new, he should be given his appointment and his instructions at an early date, so that he may be present during the period of construction and make himself familiar with every part in all its details, and learn the best methods of dismantling and assembling, the interior construction, and the arrangement of piping. When the whole is supposed to be in working order, he should make a formal and complete inspection. Should he discover anything wrong, either during construction or on later inspection, he should report it at once in order that it may be promptly corrected. If important, the report should be made in writing, and very complete; it should exhibit the nature of the defect, its probable cause and the best remedy, all expressed in as concise terms as possible. Reports on the condition of the machinery are often called for both at the beginning and at the end of the term of service, and at specified regular intervals while in operation. These reports usually include statements of all receipts and expenditures in full detail.

143. **General Instructions for Starting** the machinery into operation may be readily prescribed; but the situation may at any time demand the exercise of an independent judgment and action in defiance of general rules.

The first step is a rapid and careful inspection to make sure that all parts are in good working order. The boiler being found in proper condition, it is filled with water of as high a

temperature as can be obtained; the fires are lighted as soon as the water-level rises to a safe point; the safety-valve is lowered as soon as steam appears, all air being then expelled; and the fires are brought into good shape while the steam-pressure is rising. The time occupied in these processes ranges from a few minutes with fire-engine boilers in which the area of grate is a maximum and the weight of water a minimum, to an hour in ordinary boilers or even two hours or more in large marine boilers and where no exigency arises demanding haste. The more slowly the better for economy and safety. The person in charge should see for himself that the blow-cocks are closed before lighting the fires. When steam is up, the water-level should be brought fully up to the customary level, and kept there carefully. Both the gauge-cocks and the glass gauges should be blown out, and their certain operation insured.

Meantime the work at the engines involves the inspection of the running parts to see that they are ready and that all is clear; the adjustment of packing, of bearings, of starting gear, and of the apparatus of lubrication; the filling of all oil and tallow cups, and the opening of steam, water, and other special stop-valves preliminary to the introduction of steam and of condensing water and the discharge of the water of condensation.

Preparations for the operation of the engines and boilers should always include such inspections as are above referred to; the reparation of all defects, however small; the examination of the fits of parts, the set of keys, and the tightness of nuts; the condition of pumps, valves, packing-rings of pistons, steam and water pipes and connections, and of every part of the machinery, to see that it is sound, in good order, and properly assembled and adjusted. The safety-valves and feed arrangements of the boilers, and the valve and reversing gear of the engines, demand, particularly, the most careful examination, and they should be tried and tested until the engineer-in-charge is absolutely certain that they are properly designed and constructed, and in reliable working order.

The management of steam fire-engines and similarly high-powered machines, in which much must be sacrificed to light-

ness, and in which power, strength, compactness, and portability must be combined to the utmost possible extent, demands peculiar care and skill. The following are detailed instructions for such cases: *

I. In charging the furnace, use plenty of dry shavings and kindling-wood, filling the furnace nearly full, which in most cases will give steam enough to commence work on arrival at a fire, provided the fire is started on leaving the house.

II. In the use of coal, keep *a thin and clear fire*, the grate entirely covered. Do not break up the coal unnecessarily in the furnace. The best for this purpose is English cannel, free from dirt and dust.

III. As soon as ten pounds of steam has been generated, open gradually the steam-blower; but generally, when running, it should be kept closed.

IV. Be careful not to let fire collect under the boiler to burn the wheels.

V. Fire may be started in the boiler, with the water between the second and third gauge-cocks, the first gauge-cock being the lower one. When the engine is running, the water should generally be carried at about the third gauge-cock, which is placed near the top of the tubes. The engine may be run to its greatest capacity, by carrying the water a little lower than the third gauge-cock, which with care may be done with safety. Under no circumstances should water be carried lower than the second gauge-cock.

VI. Avoid using an unnecessary pressure of steam. From seventy to ninety pounds is as much as is generally required to do good duty.

VII. One of the feed-pumps should be worked nearly all the time when running, in order to keep the water in the boiler at the proper height, and to preserve an even pressure of steam.

VIII. When foul water is used in the boiler it is likely to produce foaming or priming. When this occurs, and it is not

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* Amoskeag.

desirable o stop the engine, the trouble may be diminished by opening the surface blower-valve, and blowing from the surface of the water the scum and oily matter which usually causes the foaming; while doing this the water should be carried as high as the surface blower-valve. When the engine can be stopped, the water should be entirely blown out of the boiler through the blow-off cock near the bottom, with a steam-pressure of about twenty pounds. Refill the boiler with fresh water, and repeat the operation until the boiler is clean.

IX. Always examine the boiler-tubes after working, and do not let them get clogged.

X. Take the engine off the springs before starting the engine, and place it on the springs again when done.

XI. See that the suction-hose and its connections are perfectly air-tight.

XII. Open the discharge-gates and cylinder drain-cocks before starting the engine.

XIII. Always start gradually. With a single long line of hose it may be necessary to open the relief-valve a little; but at other times keep it closed, except when it is desired to feed the boiler without forcing water through the hose.

XIV. The inside of cylinders, the steam-valves, the link-blocks, and all parts of the engine where there is friction, should be kept thoroughly lubricated.

XV. The main pumps should be frequently examined, and care taken that the valves and springs are in perfect order.

XVI. Every part of the engine should be thoroughly examined every time the engine has been out of the house, whether it has been worked at a fire or not.

XVII. Always keep the engine clean and at all times in perfect order in all its parts.

In the British Naval Service the Chief Engineer is expected to obtain and record the weights of all new machinery, spare gear, and supplies, with sketches or drawings of important parts, and to register all alterations.* Inspections are made

* Steam Manual for Her Majesty's Fleet.

regularly, weekly, monthly, and quarterly, in greatest detail at the longest intervals, and complete reports are made of them.

Where vessels are laid up, the pistons, rods, etc., are disconnected, or they are daily moved by hand, and are kept well lubricated. Packing is removed, and small accessory apparatus dismounted and stored. Bright parts are covered with white lead and tallow, and every loose piece suitably labelled and inventoried. Boilers are preserved, when out of use, by filling with a solution of soda, 1 part to 200 of water by weight; or, if they contain zinc slabs, with fresh clean water. The heating-surfaces and other external parts are protected against the circulation of damp air, and trays of dry lime are distributed in accessible places to take up any carbonic acid, and replaced once in six months. Stoves are sometimes used to secure dryness of all parts of the boiler and boiler-room. Cocks and valves are protected by anointing with tallow, and set tight-shut. Where danger of freezing is anticipated, the boilers are emptied. In Great Britain this is done from October to March, inclusive.

Where zinc is used, it is commonly introduced in slabs about one foot long and half as wide, and in the proportion of about one to each fifty square feet of heating-surface. About fifteen per cent of all used should be placed in the steam-space, when the boilers are filled with water. The British naval prescription for use of zinc in slabs is, in pounds,

$$z = nld \div 191,$$

where n, l, and d are the number, length, and diameter of tube.* Metallic connection must be insured.

144. Packing and Oiling are the two preliminary operations of the engine-room before commencing work with the engines. The first must usually be performed in the intervals of rest only; the latter must also be constantly and most scrupulously carried on throughout the whole working period. In the design of the engine, packing is sometimes in some places

* Steam Manual, p. 107.

dispensed with, and, when practicable, with great advantage in reducing friction and the heat of the engine-room, as well as the labor and annoyances of the attendants.

Metallic packing is much used on large piston- and valve-rods, and especially on large engines, and with good workmanship and proper care, is next best to none. Fibrous and compounded packing, containing asbestos, paper, and rubber, are more usual. They should be introduced in deep boxes, as loosely as is possible, and yet at the same time make them tight. It is best to keep the stuffing-boxes well filled, and frequent repacking will be advantageous, economically.

"Oiling up" should consist in something more than filling oil-cups. Every cup and channel should be examined and seen to be perfectly clean and clear. The quality of the oil must be carefully adjusted to the relative speeds and the pressures of the rubbing parts. Only heavy, and usually only pure, mineral oils can be used with satisfaction in the cylinder where high steam-pressures and temperatures exist. Where practicable, a free flow of oil, in steady circulation, with proper facilities for settling and straining, will give great reduction of friction and ultimate cost.*

145. **Adjusting Bearings** and "setting up brasses" is rarely practicable with any degree of nicety until some experience with the engine in hand has shown precisely how much play is allowable or necessary, and precisely how that amount may be gauged on the keys or the cap-bolts of the journal. After such experience is gained, the distance that every key or nut should be slacked up to give at once smooth action and safety against serious binding of the journal by its brasses, in case of heating, may be made a matter of record for guidance at any future time, and for the benefit of the successors of the engineer-in-charge. In heavy or fast engines, "liners" should be inserted between the brasses of the rods and other heavy parts, and so dimensioned that the latter may be set up hard upon them without danger of "seizing" the journals. As the bearing

* Friction and Lost Work; § 143, p. 332.

wears, these liners are kept adjusted by the careful use of the file.

A heated brass, if covering half the circumference of the journal, is very liable when heated to so change its shape as to grip the journal, and often with such great force as to produce heating, cutting, and sometimes the welding of the one to the other. It is advisable, for this reason, to remove a part of the bearing-surface for 15°, or 30° even, in some cases, from the line of the parting of the brasses, and this still leaves the "projected area" of the bearing but little diminished, and practically better than before. When making this disposition of the bearing-surfaces, it is well to see that a bevel is cut from near one end of each edge of each brass nearly to the other end, to catch and retain oil wiped off the journal, and also to cut a few narrow grooves to distribute the lubricant over the rubbing-surfaces between journal and brass. This is especially important where a free feed or an "oil-bath" system is desired.

Weights carried on locomotive driving-wheels have risen, since 1870, from 5 to nearly 10 tons, and on car-wheels from 3 to nearly 6 tons in heavy traffic; and these facts, as well as those elsewhere stated relative to the progress of marine engineering, give some idea of the extent to which the care of journals has come to exact the utmost care and skill in setting up and handling journals and their bearings, and in securing ample and continuous lubrication.

146. **Starting up Steam-machinery** is to be proceeded with as deliberately and cautiously as circumstances permit. The preparation above detailed being made, the engineer-in-charge ordinarily himself assumes direction, and sees the engines and boilers in smooth operation before resigning his charge to those who are to "take the watch."

The stop-valves on the boilers, and the throttle-valve on the engines, are usually opened before steam is raised, to avoid risks of their sticking or "setting fast." It has also been the experience of the Author that if a communication is thus made quite through the engines, less condensation and water-hammer are produced and less difficulty in starting, the heated air pass-

ing over before and with the steam having, often, a perceptible beneficial effect.

The engines should be turned by hand before they are moved under steam, if any general repairs have been made, to make sure that nothing has been left in the engine, and that nothing is so far out of adjustment as to interfere or to introduce danger.

On board ship the smokestack, or funnel, stays or guys are slacked off, if they have been set up when the boilers were cool, and are readjusted when steam is up.

The injection-orifices, both outboard and bilge, on the marine engine, should be looked to to see that they are not choked by weeds, chips, or other foreign substances. A steamer in shoal water or aground should be very carefully handled, especially if the bottom be of sand or gravel, lest the pumps become choked or the air-pump and condenser disabled by drawing in solid matter with the current of injection-water.

When steam is given the engines at starting, it should be by hand, for at least a sufficient number of revolutions to see all going right and to get clear of any water that may have settled in steam-pipes, valve-chests, or clearance-spaces. The relief-valves or cocks on the cylinder should not be closed until this is assured. On shipboard it is usual to get up steam well in advance of the proposed time of starting, that the engines may be worked slowly for a time to make certain that everything is in good working order.

Priming is very apt to take place at starting, and at any subsequent time if the boilers are worked hard, or a change of water occurs, or if the internal heating-surfaces are greasy. In such case the engines should be slowed down, the relief-cocks on the cylinder opened, and the fires driven less severely, or even, in serious cases, pushed back and the furnace-doors left open until the crisis is passed. After a time circulation will become well established, the foreign matters be worked off, and the machinery will then work steadily and safely.

When about to stop, the attendants in engine- and boiler-rooms should be notified as long in advance as practicable, and

the probable time and duration of the stop indicated. The fires will then be allowed to burn down somewhat, the steam-pressure and the water-level permitted to fall, and both so manipulated that a stop may be made without waste of steam or unnecessary noise at the safety-valve. When about to stop, the furnace-doors are opened, the feed-pump put on, and the fires pushed back. The engines may then be stopped without unnecessary inconvenience or expense.

While examining the engine previously to starting, the engineer should see that the valve-gear is so set that the engine cannot be started by steam leaking into the cylinders, and that the drain-cocks are open. If a locomotive, the brakes should be put on firmly and the reversing lever set at the middle notch. This should be done, also, whenever stopping to pack or to key up bearings. If keys are found loose, they should be tightened by blows with a copper- or lead-faced hammer, and to an extent measured by the position of a mark previously made on them. Loose bolts and split pins are the most frequent cause of trouble. All keys should be tightened up frequently and slightly, rather than left to pound, and then tightened up considerably.

In starting heavy engines, when cold, the greatest care must be exercised to warm up their cylinders very slowly and uniformly, as they are liable otherwise to crack, and thus give rise to enormous expense for repairs or replacement. It is a good plan to permit the air and vapor from the boilers to pass through them throughout the whole period of getting up steam. The cylinders should also be very carefully and thoroughly drained before moving the engines, and the first revolutions made as slowly as possible, especially as "the centres are turned."

Horizontal engines are to be carefully watched to detect any wearing of the cylinders—a form of wear very liable to occur with such engines. Unbalanced valves also demand even more careful watching. Loose bearings are liable to cause serious pounding, and pistons of vertical engines have been known to be split in this manner.

Rubber valves and packing are injured by grease, and its use should be avoided as far as practicable. Mineral oil is commonly advised for internal lubrication, and also mixed with a small proportion of organic oil for journals. Grease collects on condenser-tubes and reduces their efficiency. It should be removed whenever the opportunity allows by the use of a strong solution of soda. They should not be worked more than a few months at most without cleaning.

147. **While in operation,** the duties of all in engine- and fire-rooms are promptly reduced to a well-understood routine. In the engine-room little more is to be done than to keep the engine at a regular speed, to supply lubricants properly, to see that no bearings are allowed to heat, and to stop and start as required. On stationary engines the governor takes charge of the speed-regulation; with pumping-engines, and with marine engines in a smooth sea, the uniformity of the resistance insures constant speed so long as the steam-pressure is kept steady in the boiler-room; on the locomotive the speed is made, moment by moment, what may be deemed necessary to insure "running on schedule time." Lubrication consists simply in keeping the oil-cups filled, and in adjusting them to supply the oil at the desired rate.

The boilers usually demand vastly more attention than the engines.

At sea the water may contain too much salt; in which case a part of the brine is blown out of the blow-off cock and its place supplied by sea-water or by water distilled for the purpose. The saturation may be allowed to approach, but never to reach, $\frac{4}{32}$, i.e., four times the saltness of the sea, since salt begins to deposit at that point. Since all the calcium sulphate is deposited at the now usual temperatures of marine boilers, whatever the saturation, it is evidently advisable never to "blow" too freely, and hence always to keep up the saturation as nearly as is safe at that at which deposition of salt begins. Distilled feed-water is now often supplied.

The management of fires is an important but often neglected branch of instruction in fitting firemen for their special

duties. The economy of boiler management is very largely dependent upon the skilful handling of the fuel and the furnace. In general, the fires should be kept of even thickness, clear of ash and clinkers, and as clean at the sides and in the corners as elsewhere. The depth of the fuel is determined by its nature and size, and by the intensity of the draught. Hard coals can be used in greater depth than soft, and large coal in deeper fuel-beds than small. A strong draught demands a thick fire, a mild draught a thin one. With a low chimney and natural draught small anthracite or fine bituminous coal may be most successfully burned in a layer but a hand's breadth in thickness; while with large "steamboat" coal of the hardest varieties and with a heavy forced draught fires have been actually worked successfully of five times that depth, or more. The secret of success in handling fires is to find the best depth of fire for the conditions existing; to keep that thickness at all times, allowing for the ash that may accumulate; to throw the fuel on the grate at such frequent intervals as will prevent the fire burning into holes or in irregular thickness at different points: to introduce the coal so quickly and with such exactness of direction that no serious loss may occur from the inrush of cold air, and so that every shovelful should go precisely where needed, the place for the next shovelful being at the same instant located. The removal of ash is best done by means of a rake or other tool used under the grate, rather than by stirring and breaking up the bed of fuel by working through the furnace-door. The various forms of shaking-grate now in use are often very efficient. For best working, the fire should usually be kept bright beneath, and the ash-pit clear. With light draught, however, and thin fires, it is sometimes advisable, if sufficient steam can be so made, to allow the fire to be less frequently raked out, and some accumulation of ash may be thus produced when working with maximum economy.

"Firing," or "stoking," as the replenishing of the fuel is called, must be done very quickly and skilfully to avoid serious annoyance by variation of steam-pressure and supply. Where several furnaces are in use this difficulty is less likely to be met

with, as the fires may be cooled and cleaned in rotation. A skilful man will find it possible to keep steam very steadily with but two furnaces, even. Especial care should be taken to see that the sides and corners of the grate are properly attended to. Regulation of the fire is best secured by the careful, adjustment of the damper. The manipulation of the furnace-doors for this purpose is likely to cause waste. Liquid fuels are especially liable to waste by excessive air-supply, and gaseous fuel exhibits a peculiar liability to the opposite method of loss; both should be, if possible, even more carefully handled than any solid fuels.

The appearance of smoke at the chimney-top is not always indicative of serious loss, nor is its non-appearance always proof of good combustion. With soft coals and other fuels containing the hydrocarbons some smoke usually accompanies the best practically attainable conditions; anthracites, charcoal, and coke never produce true smoke. Attempts to improve the efficiency of a heat-generating apparatus by "burning the smoke" usually fail by introducing such an excess of air as to cause a loss exceeding that before experienced from the formation of smoke. Thorough intermixture of a minimum air-supply with the gases distilled from the fuel is the only means of attaining high efficiency.

Ash-pits should not be allowed to become filled with ashes, as the result would be the checking of the draught, the reduction of the steaming capacity of the boiler, and loss of efficiency, even if not the melting down of the grates. It is customary at sea to clean out the ash-pits and send up ashes, throwing them overboard once in every watch of four hours, when in full steaming. If much unburned fuel is found in the ashes, it should be, if possible, cleaned out and returned to the fire, or used elsewhere.

Cleaning fires consists in thoroughly breaking up the mass of fuel on the grate, shaking out all the ashes, quickly raking out all "clinker," as the semi-fused masses of ash and fuel are called, and, after getting a level, clean bed of good fuel, as promptly as possible covering the whole with a layer of fresh

coal. This is done, usually, once in four hours at sea and twice a day on land; but different fuels require somewhat different treatment. The work should be performed with the greatest possible thoroughness and dispatch, to avoid serious loss of steam-pressure.

Mr. C. W. Williams' instructions for handling the fires, where bituminous coal is used and an air-supply above the fuel is provided, are substantially as follows:

Charge the furnace from the bridge-end, gradually adding fuel until the dead-plate is reached and the whole grate evenly covered. Never permit the fire to get lower than four or five inches in thickness, of clear and incandescent fuel, uniformly distributed, and laid with especial care along the sides and in the corners. Any tendency to burn into holes must be checked by filling the hollows and securing a level surface. All lumps should be broken until not larger than a man's fist. Clean out the ash-pit so often that there shall be no danger of overheating the grate-bars.

An ash-pit, brightly and uniformly lighted by the fire above, indicates that it is in good order and working well. A dark or irregularly lighted ash-pit is indicative of an uncleaned and badly working fire. The cleaning of the fire is best done, in ordinary working, by a "rake" or other tool working on the under side of the grates, and not by a "slice-bar" driven into the mass of fuel and above the grate.

Different fuels require different treatment. The principles just stated apply generally, but more, perhaps, to anthracite coals. The soft coals are commonly so disposed on the fire that a charge may have time to coke and its gases to burn before it is spread over the grate; liquid fuels must be so supplied that they may burn completely, at a perfectly uniform rate, and especially in such manner as to be safe from explosive combustion, the same precaution is demanded with the gaseous fuels. Special arrangements of grate and a special routine in working may be, and often are, demanded in such cases.

The liquid and gaseous fuels are often and successfully

burned in conjunction with solid fuels. In such cases the same methods are to be adopted and precautions observed in handling the latter as when burned alone.

The liquid fuels are almost invariably the crude petroleums. They are sometimes burned in a furnace in which they are allowed to drip from shelf to shelf in a series arranged vertically at the front of the furnace, the flame passing to the rear, with the entering current of air supporting their combustion. In many cases they are sprayed into the furnace by a jet of steam, which should be superheated, and at high pressure. The use of the steam is considered to have a peculiar and beneficial effect, possibly through chemical reactions facilitating the formation of hydrocarbons. The petroleums are all liable to cause accident if carelessly handled, and special precaution must be observed in their application to the production of steam.

The gaseous fuels are seldom used under steam-boilers, except where "natural" gas from gas-wells is obtainable, or where a very large demand or the use of metallurgical processes justifies the construction of gas-generators. Even greater precautions against accidents by explosion are needed than with the liquid fuels. In burning gas, maximum economy is secured by careful apportionment of the air-supply to the gas-consumption, and especially in avoiding excess. The regenerator system is not generally economically applicable to boilers.

Good furnace management, to secure maximum heat-supply from the unit weight of fuel, is evidently as essential to economy and efficiency of steam production as choice of proper fuels.

The smaller the coal, where anthracite is used, the thinner should be the fire; the stronger the draught, the thicker the bed of fuel, of whatever kind. With too thin a fire the danger arises of excess of air-supply; with too heavy a fire, carbon monoxide (carbonic oxide) may be produced. In the former case combustion will be complete, but the heat generated will be distributed throughout the diluting excess of

air, and thus rendered less available, and the efficiency of the furnace will be correspondingly reduced; while in the latter case a loss arises from incomplete combustion, and waste takes place by the passage of combustible gas up the chimney. The second is the less common cause of loss of the two, but both are liable to arise in almost any boiler, and we may even have both losses exhibited in the same boiler and at the same time. Successful working demands a very perfect mixture of the combustible with the supporter of combustion, and should this not be secured, serious waste will take place.

In the management of the furnace the effort should be made to secure the best conditions for economy, and as nearly as possible perfect uniformity of those conditions.

In handling the locomotive engine the steam is given full-stroke at starting, and the reversing-lever gradually taken up toward mid-gear, as full speed is approached, until the proper cut-off and best working conditions for the load are reached. In starting and stopping, the throttle-valve should be opened or closed when the reversing-lever is in mid-gear.* When at full-speed, the steam should be at maximum pressure and steady. When it is known when the start or a stop is to take place, it is always perfectly easy to prepare for it at the boiler; but a start without ample warning or a stop in an emergency may cause some trouble, and often only a skilful and experienced engineer can evade serious risks.

The operation and management of the multi-cylinder engine are mainly identical in principle and in practice with the methods found best with the simple engine. Greater care is needed in keeping the intended pressure of steam and its full supply, in securing absolutely dry steam, and in preserving the correct *régime* in operation, than with the older machine. This system is not as elastic as is the older one, and is, for this reason especially, peculiarly suitable for engines doing invariable work, as in long-voyage steamers. This type of engine is well calculated especially for use where all the requirements

* For details, see Forney's Catechism of the Locomotive and Sinclair's Locomotive Engine-running; N. Y., J. Wiley & Sons.

and conditions of economy assume constant and considerable importance.

The jackets should be carefully kept in working order, and especially free from air or water.

In the operation of multiple-cylinder engines the necessity should be constantly in view of ascertaining the best ratios of expansion for each cylinder to properly divide the work, under the various conditions of their customary use, as well as those giving highest economy of steam and fuel. When these conditions are determined, the results of these observations should be put on record as a permanent guide.

On all engines, and in all parts of engines, boilers, and hull, voltaic action should be very carefully watched for and guarded against. It is especially dangerous in boilers, and is peculiarly liable to occur with surface condensation, and wherever brass, bronze, or copper in masses are in metallic connection with wrought-iron or soft steel.

Hard-driven machinery, as that of torpedo-boats, demands the utmost care and vigilance.

Where superheaters are used, they are, where possible, to be shut off whenever the engines are not drawing steam; and water should not be allowed to accumulate in them.

Since the thermal wastes increase rapidly with increase of ratio of expansion, it may sometimes be desirable, in the case of an underloaded engine, where choice is permitted, to apportion the work to the load by throttling the steam rather than by expanding it from the boiler-pressure. This is probably a rare case in practice, however, and it is easy to show that in general the use of the "reverse-lever" on the locomotive, as an example, is preferable to using the throttle-valve. Thus, taking a pressure of 180 pounds by gauge, 6 pounds back-pressure, or 195 and 21, respectively, very nearly, above vacuum; feed-water at 60° F., ratio of expansion constant at 4 in the one case and variable in the other; expansion nearly adiabatic, as assumed by Rankine; and compression sufficient to permit the clearance-wastes to be ignored.*

* These cases were computed by Mr. H. W. Hibbard.

These computations, made as usual and already illustrated, indicate an ideal thermodynamic efficiency for maximum pressures and the specified ratio of expansion of 13.2 per cent, ranging rapidly downward to 7.8 per cent at a pressure of 60 pounds by gauge; thus:

r	p_1	E
4	195	13.2
"	175	12.7
"	155	12.2
"	135	11.7
"	115	10.9
"	75	7.8

With a constant maximum pressure, the ratio of expansion ranging between 9 and unity, the efficiency varies from 13.9 down to 6.4 per cent, and falls off again above $r = 7$, the gain being rapid between $r = 1$ and $r = 4$, slowing quickly between the latter figure and 7, and then loss occurring and a slow decrease of efficiency; thus:

p_1	r	p_e	E'
1.95	1	174	6.4
"	2	141	10.4
"	4	90	13.2
"	6	63	14.0
"	8	47.5	13.9
"	10	37	13.7

The mean pressure at $r = 4$ is

$$p_m = 0.570 p_1;$$

and, taking this constant, assuming a momentarily constant load, the two methods of securing this value may be compared.

It will be found that the efficiency with throttling to the required initial pressure and constant $r = 4$, will be thus:

p_1	p_m	p_e	Effic.
284	162	141	....
195	111	90	13.2
148	84	63	12.0
120	68.5	47.5	11.1
102	58	37	10.1

The efficiencies while throttling are not far from

$$E = 1.6 \sqrt{p_e}.$$

At the same time, for similar values of p, here taken in pounds on the square inch, the ratio of expansion varying, we obtain, as already given:

p_1	r	p_m	p_e	Effic.
195	2	162	141	10.4
"	4	111	90	13.2
"	6	84	63	14.0
"	8	68.5	47.5	13.9
"	10	58	37	13.7

These last values are all between 13 and 14 per cent for practicable cases, and average 13 per cent. Thus the ratio

$$\frac{E'}{E} = \frac{13}{1.5 \sqrt{p_e}} = \frac{9}{\sqrt{p_e}}, \text{ nearly,}$$

which usually exceeds unity; and throttling seems the more advantageous as compared with an expansion-ratio less than 4; but expansion is preferable with mean pressures less than 90, corresponding to $r = 4$, and the advantage increases with increasing expansions and lighter loads.

These ideal conditions are modified in practice by the considerable and variable internal wastes which are exaggerated rapidly with high ratios of expansion, and by the fact that throttling may insure some gain by drying the steam or even superheating it. Both facts tell against expansion, and favor superheating. Experience indicates that, with usual loads and speeds, the advantage still lies on the side of the use of the reversing-lever rather than of the throttle-valve, especially where reduced resistances occur with increasing speeds, and thus an amelioration of the internal losses. It is very probable that in many cases moderate throttling may prove useful simultaneously with increasing expansion.

148. Accidents and Casualties resulting seriously occur from innumerable causes, and the most thoughtful and far-seeing engineer will sometimes find himself at fault in an emergency; but it is nevertheless his duty to endeavor to think out, and to have ready in his mind, a remedy for every disaster that he can foresee, and a system of prompt action in every emergency. When the moment arrives, he will then be very likely to act instantly and intuitively on the plan previously conceived, and may thus save life and property which hesitation, even for a moment, might sacrifice.* Every good engineer has a mind stored with such plans.

The most usual causes of serious accident are water entering the cylinder, priming from the boiler when the latter is hard driven, and the breakage of a crank-pin. The first may initiate the second, or may produce fracture of any or all the running parts of the engine; or even, as has not infrequently happened, the wrecking of the whole engine—cylinder, frame, condensing system, and all. Any breaking in the line from piston to crank-shaft may precipitate such a catastrophe. The

* The Author was once standing behind the horizontal 30-inch cylinder-head of a U. S. naval vessel, with his hand on the "throttle," the engine making 100 revolutions a minute, going into action, when the cross-head broke. The valve was intuitively closed, and before the engine could take steam on the other side. No further damage was done. Had the head been knocked out, several lives might have been lost.

fracture of a crank-pin or of the shaft is usually due to the extending of some defect, at first unobserved or apparently insignificant, by the continual reversal of the stresses and by the unceasing shocks to which those parts are more or less subject.

A flaw is hardly less certain to gradually extend in iron or steel than in glass. Its first effect is to concentrate the strain in its own neighborhood, and to thus correspondingly reduce that resilience which is an essential element of security in such machinery. It thus is extended at every impulse until, unequal to even its regular work, it breaks; and often the fractured surfaces exhibit plainly the successive steps of the whole process.* A very large proportion of serious and avoidable accidents, aside from the class first mentioned, are a consequence of such progressive weakening, until the factor of safety is reduced to its limit, unity. In many cases vigilance and conscientious inspection at proper intervals would prevent accident. In some cases, however, the construction of the engine may be such as to conceal a threatening disaster.† The designer is in such cases responsible, and, next to the designer, the constructor. According to Mr. Longridge, averages of break-downs may be classified as below: ‡

37 per cent due to causes purely accidental or unascertained.
18 " " " negligence of owners or attendants.
14 " " " old defects and wear and tear.
31 " " " weakness from faulty design or bad workmanship.

The particular part of the engine to which the damage in each case seemed to have been due, or, where the damage was due to something external to the engine, the part which seemed to have broken first, is given in the following list:

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* A crank-pin sent the Author by Mr. Wm. Hewitt shows a gradual yielding and a constant polishing of the breaking-surfaces, where opposed, until only about one third the total section was left at the final crash; the rough, freshly-broken area plainly measuring its extent.

† In the case just described, the flaw was concealed by a counterbore in the crank which received that part of the pin.

‡ Engineering Journal, November, 1890

	During 1889.	Numbers during previous 9 years.	Totals.
Spur-gearing	20	214	234
Valves and valve-gear	25	211	236
Air-pump motions	13	140	153
Air-pump, buckets, and valves	8	92	100
Columns, entablatures, bed-plates, and pedestals	11	67	78
Bolts, screws, gibs, cutters, and straps	15	67	82
Main shafts	7	58	65
Parallel motions, links, and guides	8	57	65
Pistons	3	35	38
Cylinders, valve-chests, and covers	3	29	32
Fly-wheels	2	26	28
Piston-rod cross-heads	6	25	31
Piston-rods	4	23	27
Cranks	1	21	22
Governor-gear	1	20	21
Air-pumps and condensers	4	18	22
Crank-pins	1	12	13
Gudgeons in beams	2	11	13
Beams and side-levers	0	9	9
Connecting-rods	1	8	9
Total wrecks, causes unknown	0	5	5
Second-motion shafts	0	1	1
Main driving-ropes	1	0	1
	136	1149	1285

Minor accidents to machinery on shipboard can on war-vessels generally, and on merchant-steamers often, be remedied by the use of facilities on board.

Repairs of the larger parts and of portions of the engine exposed to great strains must usually be deferred until the vessel arrives in a port where suitable repair-shops may be found; but occasionally even these casualties may be tempo-

rarily repaired, and the ship enabled to proceed to port under low steam.

Professional knowledge and ingenuity will often conquer obstacles that at first appear absolutely insurmountable. On land, the proximity of the shops renders such talent less essential.

The steam-cylinders often become injured by what is technically termed "cutting," and are sometimes cracked or broken by the entrance of water with the steam (foaming), or other causes.

"Cutting" is the term applied to the formation of longitudinal grooves on the interior surface, and may be produced by setting out the piston-rings with too great force, or by foreign material entering the cylinder and being ground between it and the piston. The "cutting" of the cylinder is indicated by a peculiar sound produced by every stroke of the piston. If the metal of the cylinder is much softer than that of the piston-rings, cutting is always liable to occur.

If the furrows produced by either of the above causes are very shallow, they should be removed by the file or the scraper in the hands of a careful workman, great care being taken to retain the true cylindrical form of the surface.

If the injury is too extensive to be repaired by the scraper or file, the attempt may be made to fill the grooves with Babbitt or type metal; and if still unsuccessful, they should be cut very smoothly, and strips of metal fitted in and filed down very exactly to surface, with the aid of a template of precisely the same radius as the cylinder.

Such injuries have sometimes been repaired, where piston-rings of steel could be made, by giving the rings as good an edge as possible, and after properly hardening them, replacing them, setting them out carefully, and turning the engine by steam until the rings have *planed* out the cylinders properly. This expedient is somewhat hazardous, but with care may meet with perfect success. After repairing both cylinder and piston-rings the latter should be restored to their places, but

turned somewhat from their original positions, in order that the two depressed parts may not be in juxtaposition.

Cracked or broken cylinders are generally caused by water obstructing the movement of the piston when near the end of the stroke; sometimes by flaws in the casting, or by blows from heavy bodies falling against them; exceptionally, by having strength insufficient for safety in regular work; and by strains caused by irregular contraction of the casting while cooling in the mould.

When a crack is limited in extent, but it is feared that it may extend, it may often be arrested by drilling a hole at each end and filling the hole by a screw or a rivet. This probably prevents the lengthening of the crack by distributing the strain over a greater section. If the crack allows a leakage of steam and water, holes should be bored side by side throughout its length, and copper or other soft-metal screws or rivets should be driven into them until they are found to become steam-tight.

On the inside of the cylinder the holes should be countersunk and the rivet-heads cut down level with the surface. Being of soft metal, the passage of the piston across them will rub down their heads until a nearly perfect surface is obtained.

If the crack is too large, or the cylinder too much weakened for the method just described, a "patch" must be applied.

This may be either the ordinary patch of boiler-iron, secured by bolts and rendered tight by a putty of red and white lead, or a patch fitted in similar manner, but with a cement of iron-filings; or, where it will not interfere with the movement of the piston, two plates should be used, one inside and one on the outside, held in place by the same bolts.

When the fracture is of considerable size, and extends nearly or exactly lengthwise the cylinder, the latter may be strengthened by circular bands of iron—usually about $\frac{1}{2}$ or $\frac{3}{4}$ inch in thickness, and three or four inches wide; and when it becomes necessary to apply these to portions of the cylinder not strictly cylindrical, as at the valve-chests, they may be carried over the elevated portions, and the open spaces be-

tween band and cylinder filled in firmly with blocks of wood; or the bands may be tap-bolted to the cylinder at each extremity, as near the obstruction as possible.

When the fracture extends around the cylinder, the crevice should first be rendered perfectly steam-tight in the manner already described, and then, if the cylinder is seriously weakened, bolts should be carried from top to bottom, and so arranged as to draw and hold the two parts firmly together.

A bent rod can sometimes be straightened by heating in a wood or, better, charcoal fire, great care being taken to keep it well under the fuel, to prevent oxidation, and then forcing into shape by the blows of a heavy wooden hammer, or a maul, which may usually, if necessary, be improvised for the occasion. A crank-pin must be treated with especial care.

A broken cylinder-head may sometimes be patched; if not, or if it cannot be reinforced safely, it may be practicable to remove it and to block up the ports at that end and work the engine "single-acting." There are few results of accident absolutely without either remedy or evasion.

A broken pin, or even shaft, may often be roughly repaired and worked at reduced power until permanent repair can be effected.

Worn brasses should be carefully readjusted, and especial care should be taken in setting up those in the connecting-rod ends to see that the piston be not drawn toward the end of the cylinder in such manner as to seriously reduce the clearance-space. Worn brasses should be replaced before wear has weakened them, and thus danger of their fracture incurred.

Accidents in the boiler-room are commonly the blowing out of a stay-bolt, the bursting or splitting or serious leakage of a tube, and, on rare occasions, the "coming down" of a furnace-crown or the fracture of an area or line of weakened plate in the shell of the boiler. If, in either case, steam issues into the room, the safety-valve must be instantly raised, the furnace-doors opened, the feed put on in maximum volume, and where practicable, as at sea, the engine worked at maximum speed with open throttle-valve and link down to give

steam as nearly as possible full-stroke, until the issue of steam has been reduced so as to remove all danger. If there are more than one boiler, that injured is shut off from the rest as quickly as practicable and repaired. Burst "fire-tubes" can often be plugged easily and quickly without even extinguishing the fires or blowing down; but water-tubes must always be emptied, and the resulting delay accepted as inevitable. Parts of the boiler requiring it may often be perfectly well repaired with a "soft patch" of carefully fitted iron bolted in place, and the joint made tight by a cement of red and white lead, or of red lead, oil, and iron-borings. A sheet removed or a large area of shell being dangerously thin, a "hard patch," riveted in place, with well-calked seams, is the only proper remedy.

Cracked steam-pipes and leaky seams may often be made tight by a wrapping of canvas or sheet rubber smeared with red and white lead or other cement, and closely wound with heavy tarred cord or twine.

Leaky valves must be ground in or replaced and the feed check-valves on the boiler, and the blow-off cocks should be especially looked after.

In some cases accidents to machinery are at the moment irremediable, and the only course is to get the aid of the builder as promptly as possible; in other instances, in many minor cases of accident, some remedy can be devised by the competent engineer, that will at least carry him safely to a time when repairs may be made without trouble. Where a pair of engines is employed, as on shipboard, and with locomotives, it is usually possible to get on with one engine, even though the other is completely disabled. Broken parts may sometimes be at once replaced by spare duplicates, which a careful engineer likes to have on hand for the more important and more dangerous pieces.

Heated journals should be cooled, if with cold water, very slowly and cautiously; pillow-blocks have sometimes been split by too sudden cooling. The Author prefers to use sulphur and oil as a cooling fluid for bad cases.

Injured men should be instantly placed in the hands of a

skilled surgeon or the nearest physician. Not a moment should be lost, as in a bad case the delay may have serious results. In many cases even immediate treatment only may save life—as where an artery is injured and bleeding freely. A badly injured man may die of loss of blood, of simple pain, or fright, or, even if not otherwise dangerously hurt, of shock or collapse.

When suffering from shock, he is cold, pale, and faint, if not insensible, and his breathing is difficult and restricted. He should be at once given a stimulant, unless it has already been used by the patient as a beverage, and warmth applied to the body in any immediately available manner. When bleeding, the artery must be found and taken up; meantime, bind the part snugly on the side toward the heart to check the flow until some one understanding the case can take it in hand. The artery can usually be discovered by its pulsations, and a compress or an improvised pad bound over it to stop the current. If in arm or leg, a handkerchief tied closely about it and converted into a tourniquet by a stick inserted so that it may be used to twist it, will answer the purpose. No more pressure should be applied than is just necessary to stop the bleeding.

In carrying the victim a soft and even bed should be made up for him, and the greatest care and gentleness employed in his removal.

Burns or scalds, such as are liable to occur where steam is used, if not extensive and if the skin be not actually ruptured, may be treated by softly bandaging with cooling lotions, or lightly covering with fine, soft linen anointed with a soothing salve, as "simple cerate." If the true skin is exposed, dress with fine flour, starch, sweet-oil, or other non-irritating substance that may exclude the air and yet not heat the part. *Cold* applications should not be applied; warm water is better, safer, and more agreeable.

Suffocated men should be treated as if just rescued from drowning. Insensibility from overheating is treated as sunstroke, the patient being bathed about the head, neck, and face with cold water in a comfortable but not cold locality.

149. Wear and Corrosion occur with both engines and boilers: either may ultimately lead to accidents; but they progress so gradually that there is seldom any excuse for such a result. In the case of the boiler, where heat, steam, carbonic gases, water, and other unfavorable conditions conspire, corrosion is the great enemy, and often produces rapid decay and resultant expense and danger. At the engine, the wear of rubbing parts is the cause of depreciation. The boiler usually has a comparatively short life; the engine, if well cared for, a very long one. Boilers have been ruined by corrosion in a few months, and seldom last many years; engines meeting with no accident resulting in a general wreck should endure a century, and are usually only displaced by later types.

Wear is made a minimum by insuring ample and never-interrupted lubrication, and, the area of rubbing parts having been made ample by the designer, it should long remain imperceptible. All parts exposed to wear are made readily removable and easy of replacement.

Corrosion at the boiler is checked internally by providing against the use of water free from acid or salts, and if it has or is given a slightly alkaline reaction, so much the better. Externally the boiler surfaces must be kept dry, and every incipient leak checked. Contact with the brick and mortar of a moist setting or the erection of the boiler in a damp place will be certain to prove expensive and troublesome, if not dangerous.

Every part of the boiler should be accessible, inside and out, so that periodical inspection may lead to the discovery and prompt remedy of the first steps of this process.

When the feed-water is found liable to carry into the boiler organic and oily material, and to produce the peculiarly dangerous corrosions or the deposits so frequently caused by them, it should be carefully filtered through a coke-column, or the best mineral oils should be used if practicable, thus to evade danger. "Scum-cocks" blowing out surface water should be employed in such cases, and even considerable expense may be often justified in changing the source or the composition of the feed-water and its contents. Even brackish or salt water is prefer-

able to an oily or mucilaginous supply, as the dissolved salts may be kept below the objectionable proportion by proper blowing off, and the scale, if formed, is much less dangerous.

Wear and corrosion of parts goes on rapidly about the boiler, even with the best of care, and its life is but a few years at most, if doing hard work, especially at sea or in damp situations on land; but with proper care the engine should, as already remarked, last almost indefinitely. The Author has known a main-journal brass to work ten years constantly, in a marine engine, without appreciable or measurable wear. The wear of well-proportioned journals is usually the measure of the inefficiency of their lubrication.* The same is true of piston-rings, but less commonly of valves and their seats, which are more difficult, though even more important, of lubrication. Wear of parts should be taken up as soon as noticeable.

150. **General and Special Repairs** are to be made so often as to secure prompt remedy of any matter going wrong. The proverbial "stitch in time" is especially to be enjoined upon those who have charge of steam-machinery. Stationary engines and boilers are taken in hand at such times as will least impede business, and at such intervals as may prove by experience to be on the whole most economical. Locomotives may be overhauled after each run, and may usually be taken off the road, whenever it may seem advisable, without great inconvenience. Steam-vessels are overhauled at every return to port at the home-end of the route.

Whenever extensive special repairs are rendered necessary by accident or exceptional circumstances, the opportunity is taken advantage of to make a general inspection, and all other desirable, even if not immediately needed, repairs.

In many cases general repairs are made at intervals and in a manner specified by general standing instructions to those charged with their conduct. In such instances old parts are

* See Friction and Lost Work, etc. (R. H. Thurston; N. Y., J. Wiley & Sons), for proportions of journals and methods of management.

sometimes replaced by new, even though no evidence of immediate necessity appears. Thus, the axles are removed from under passenger-trains and placed beneath freight-trains after specified periods and before the slightest danger can be ordinarily apprehended; steam-boilers are given complete annual and regular quarterly examination by insurance inspectors; locomotives must pass under the eye of the master-mechanic at stated times, even though presumed by those in immediate charge to be in perfect order. Where, as in such cases, life and property are at perpetual risk, the precautions taken are commensurately minute and exacting.

151. **The Replacement of Parts** is rarely required in engines, unless as the result of accident. All the wearing parts are, or should be, of ample area and well lubricated, and wear is exceedingly slow so long as lubrication is kept up properly. The failure of the oil-supply, leading to the "cutting" and destruction of brasses, is the most common source of this expense. Spare brasses already fitted should be kept on hand for immediate use in the more troublesome bearings, as the crank-pin, cross-head pin, and main bearings, to avoid the serious consequences liable to follow delay in replacement.

Rods and connections, especially with slide-valve engines, about the valve-motion may be overstrained and even broken, and spares are often provided for such parts. For example, the British "Lloyd's" directs that every marine engine have stores as follow: 3 connecting-rod top-end bolts and nuts; 2 connecting-rod bottom-end bolts and nuts; 2 main-bearing bolts; 1 set coupling-bolts; 1 set pump-valves; 1 set piston-springs; a supply of assorted bolts and nuts, iron and steel of various kinds.

It is also advised that the following spare parts be carried: 1 crank-shaft; 1 propeller-shaft; 1 propeller or set of blades; 1 stern-bush or lignum-vitæ lining; 1 air-pump rod; 1 set valve-spindles; 1 circulating-rod; 1 set check-valves; 1 set rod-brasses; 1 eccentric-strap; 1 set link-brasses; 6 junk-ring bolts; 2 dozen boiler-tubes; 6 cylinder-cover bolts; 3 dozen

condenser-tubes; 4 valve-chest bolts; 1 set safety-valve springs.

It is usually advisable to work night and day in repairing or replacing parts.

152. **The Inspection** of machinery at intervals is a proceeding evidently advisable as the only means of keeping track of deterioration or of detecting some classes of progressive failure. Marine engines and boilers are commonly subject to government inspection. The following are the British Lloyd's rules on this head: *

The machinery and boilers of all steam-ships are to be surveyed annually, if practicable, and in addition to be submitted to a special survey every four years, and the boilers to special survey when six years old, and subsequently to annual survey.

At these special surveys the propeller, stern-bush, and fastenings of the sea connections are to be examined while the vessel is in dry-dock, and, if deemed necessary by the surveyor, the stern-shaft is to be drawn and examined.

The cylinders, pistons, slide-valves, crank-shaft, and pumps are to be examined, and if necessary the condenser is to be examined and tested.

The boilers and superheaters are to be examined, and if deemed necessary by the surveyors are to be drilled or tested by hydraulic pressure; the safe working pressure is to be determined by their actual condition.

The safety-valves are to be examined and set to the safe working pressure.

The sea connection and arrangements of cocks, pipes, bilge-suctions, roses, etc., are to be examined.

For his own satisfaction, however, every good engineer, whenever the opportunity offers, makes inspections far more complete and fruitful of information than the regular stated surveys.

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* Lloyd's rules; § 78; 1888.

153. The "Laying-off" and Preservation of the machinery is a matter demanding both experience and judgment. When the further use of the apparatus is uncertain as to time, and when, especially, it is probable that it will be necessary to preserve them, out of use, indefinitely, it is important that the engineer should make certain that no surfaces exposed to the atmosphere are unprotected, by grease or other preservatives, from corrosion. A mixture of white-lead and tallow, hard enough to withstand slight rubbing, is commonly used to cover brightly finished parts; paints are employed on the unfinished surfaces, and small interior spaces are filled with oil or tallow. Large interiors, as of the boilers, may be protected by fish-oil, or closed up full of slightly alkaline water. The presence of caustic lime or other alkali in enclosed spaces will give indefinite insurance against oxidation, since moisture and carbonic acid must both be present to effect it.

When the machinery is to be kept in readiness for use, or laid off for a probably short time, it is not necessary to do more; but if it is to be laid aside very long, it is better to dismantle it entirely, protecting exposed surfaces as above indicated, wrapping all journals with canvas or a layer or two of heavy cord or rattlin stuff closely laid on, and storing it all in dry localities. Each piece should be marked for identification, and all boxes so tagged as to make it easy to ascertain the whereabouts of any piece so cared for, and the contents of every package.

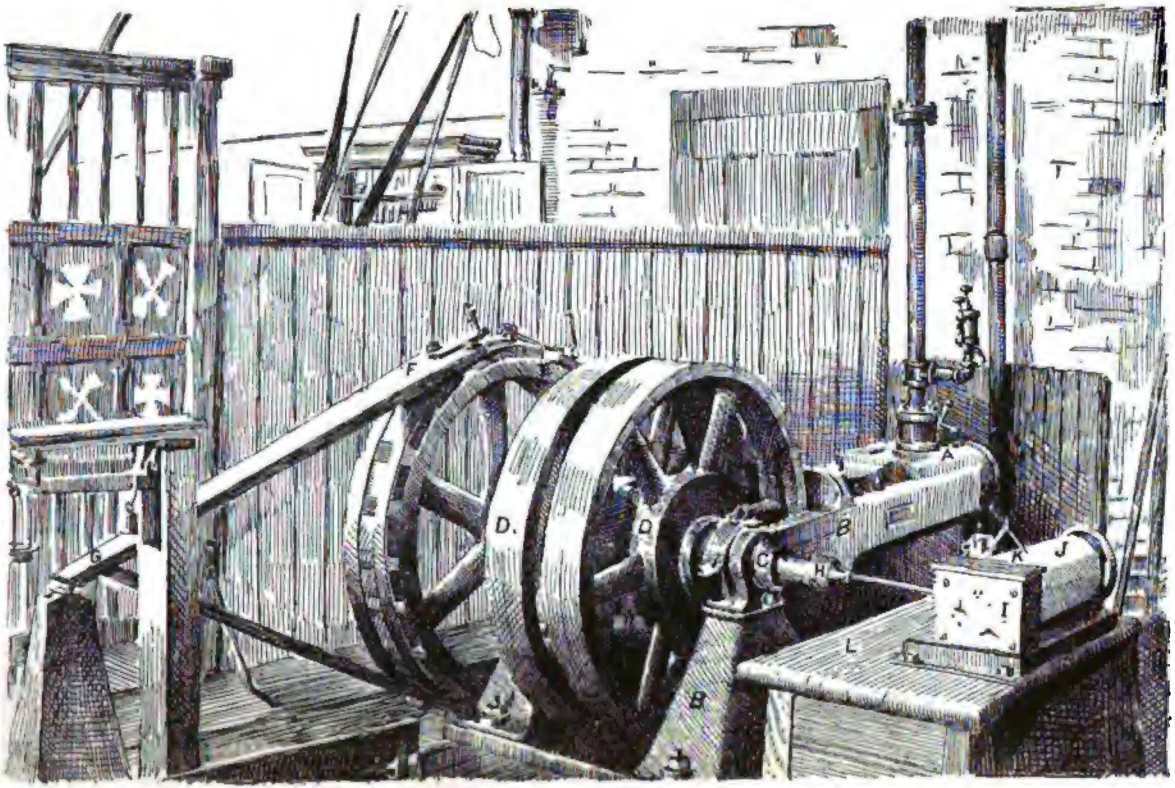

FIG. 188.—ENGINE TESTING. CHRONOGRAPH ATTACHED. (§ 205.)

CHAPTER VI.

ENGINE AND BOILER TRIALS.*

154. **The Purpose of Test-trials** of engines and boilers is, commonly, the verification of the claims of the builder to complete fulfilment of his contract, and more especially as to the power and economical working of his apparatus. Whenever a motor, whether an air, a gas, or a steam or other vapor engine, is constructed for a proposing purchaser and user, the builder is expected to bind himself by a carefully drawn contract to supply apparatus capable of developing a stated amount of power and with a specified consumption of fuel, and, sometimes, of other supplies. A test-trial is demanded, when the machine is set up and in normal operation, to ascertain whether such contract and its specifications have been completely fulfilled.

In other cases a trial is made to satisfy the proprietor that his machinery is doing good work; in still other instances he desires to ascertain whether variations of the usual methods of operation and rules of management may be expected to give improved results; sometimes he desires to test the skill of his men, or the character of the fuel employed. In all cases, whatever the main purpose of the operation, certain data are sought to be obtained as a basis for computation of the results needed, either to give a measure of the power and efficiency of the machinery or a means of comparison with other apparatus of similar character and known excellence.

Scientific Investigations require the collection of considerably more data. This object, as stated by Professor Carpenter, is to study the action of the steam, as it successively passes

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* Largely condensed from "A Handbook of Engine and Boiler Trials;" R. H. Thurston; N. Y., J. Wiley & Sons.

each portion of the cylinder; to ascertain the loss or gain of heat thermodynamically in each part of the stroke; and to find the relations of this heat-waste to the energy supplied.

Such tests are made by either of two methods; the first of which establishes thermodynamical relations from the measurements of pressure and volume as exhibited on the indicator-diagram. In this case the form of the card becomes of great importance, as loss or gain of heat in every portion of the stroke is to be determined by the positions of successive points, which are to be compared with corresponding points on a curve having known properties. In this case it is necessary to know in advance the thermodynamical relations of pressure and volume. The second method was devised by M. Hirn, and was termed by him the practical method. In this case the form of the curve is entirely disregarded, the only important data determined being the external work done from point to point, and the relative volumes of working fluid, at important events, during the stroke. He considers the engine as receiving a certain amount of heat at admission; which is reduced by external work during admission, and by loss in heating the cylinder, and is increased by the heat in the steam already existing in the clearance-spaces. This must equal that in the engine at cut-off. This, again, reduced for external work done and cylinder absorption, must equal that which exists in the cylinder at exhaust. The amount of heat thrown out by the exhaust is to be measured independently, and serves as a check on results. The difference between this heat and that at the beginning of exhaust, reduced for external work and cylinder loss, is the heat remaining at compression.

If we measure the quality and amount of steam as it enters the cylinder, and the quality of the steam during exhaust and compression, we have seen that we can establish five equations into which will enter as unknown quantities, the effect of the metal absorbing or giving out heat for each portion of the stroke. These equations establish the relation of cylinder-waste to heat supplied, for each portion of the stroke, and are of great value in determining the efficiency of the engine. The

methods of measurement and computation of the required quantities will be here briefly outlined, and forms for use in engine-trials and for Hirn's analysis are given in the Appendix.

In the naval service trials of engines are made on various occasions,—as the general trial to determine the condition of the machinery; preliminary trials to see all right; contractor's trials to show that the terms of the builder's contract have been complied with; commissioning trials to ascertain if all is ready for the ship to go into commission; repair trials to ascertain if all repairs have been properly made; ordinary measured-mile trials to ascertain speed; and the special speed trials to find the most economical speed and the best speed at full steam-power. In making deductions from the indicator-diagrams in such trials, it is customary to take the product of the total pressure at the termination of the expansion-line by the revolutions in the unit of time as a measure of the coal-consumption, which usually gives a fairly approximate result, where the conditions of wind and sea remain constant.

A complete trial of engine and boiler involves the determination of the quantity of energy stored in potential form in the fuel; the amount liberated by combustion in available form; the proportion and the quantity taken up by the boiler; the amount stored in the steam, and in any water taken up by it, and transferred to the engine; and the distribution at the engine into useful and lost work, and wasted heat. The methods of computation of these quantities will be given presently.

The purposes and methods of such trials are thus the exact and unquestionable determination of one or several of the efficiencies of the engine—or the boiler—and these methods are usually intended to be such as will give scientifically accurate measures of the heat, the steam, the feed-water, and the energy supplied to the system; the heat, steam, and energy reaching the engine; the power developed; the distribution, usefully and wastefully, of heat, energy, or power, or of all; the power and the thermodynamic and the actual efficiency of the engine considered as a heat-engine; and also the efficiency of

the engine considered as a train of mechanism, i.e. as a machine. It is not always essential that all these determinations shall be made, or that such as are made shall be rigidly exact. Trials are often made which give partial results, and by methods which are only approximate, and sometimes but roughly approximate, if judged from the standpoint of experimental science. As in all engineering work, the ultimate gauge of expediency, as judged from a financial standpoint and with an eye to a final summation of results, determines the extent to which the engineer is justified in giving his time and incurring expense in making steam-engine and boiler trials. On extensive contracts and important and costly work all the resources of physical science and engineering practice are applied; in minor matters, but little expense or labor is deemed justifiable.

155. **Specifications of Performance,** and, often, a guarantee, with forfeiture in case of non-fulfilment, should form a part of the contract; and those assurances of efficiency should be so exact and definite that no question can arise as to their meaning and fair interpretation when the time arrives for their verification. The customary forms of such specifications are now fairly well settled, and the usual methods of comparison and verification will be exhibited and exemplified. When no such specification exists, it is assumed that the maker is bound to do reasonably good work and to assure to the buyers reasonably good economical performance. Obvious and unquestionable delinquency, as shown by test-trial, relieves the buyer of every responsibility not specifically and unqualifiedly assumed, and throws it upon the constructor and vendor.

Engine Duty is commonly the technical measure of the efficiency of the engine as determined by the cost of the work done in fuel consumed. The "horse-power," taken in British measure as 33,000 foot-pounds per minute or 1,980,000 per hour, demands the transformation of the equivalent amount of heat into work each minute or hour; which quantity should be supplied by one fourth of a pound of good fuel, or less than 2¼ pounds of steam, as worked in the perfect, ideal engine having an efficiency unity. The actual consumption of en-

ergy derived from the boiler, as will be seen later, is rarely less than eight or ten times these amounts.

Pumping-engines are commonly rated by the work done by the consumption of a specified weight of fuel, as one hundred pounds. A duty of 100,000,000 foot-pounds, on this basis, would correspond to a consumption of 1.98 pounds of fuel per horse-power per hour.

Mr. Emery has compared steam-engines of various kinds on the assumption that the boiler is capable of absorbing 10,000 heat-units per pound of coal consumed. This corresponds to an evaporation of 8.99 pounds of water at 80 pounds, 9.03 pounds at 60 pounds, or 9.08 pounds at 40 pounds gauge-pressure from a temperature of 100° F. in each case. Ten thousand heat-units per pound of coal is equivalent to one million heat-units per 100 pounds of coal; and as the duty of pumping-engines is conventionally expressed in millions of foot-pounds per 100 pounds of coal, it follows on the basis presented that *the number of foot-pounds per heat-unit represents also the number of millions of foot-pounds duty per* 100 *pounds of coal.** The performance of all kinds of steam-engines may be readily compared on this basis. Ten thousand heat-units per pound of coal represent an efficiency of only ($10,000 \times 100 \div 14,500 =$) 69 per cent of the calorific value of pure carbon and of the best of fuels; so that ordinarily more than ($100 - 69 =$) 31 per cent of the heat in the fuel is carried up the chimney. The mechanical equivalent of one heat-unit is 772 foot-pounds, which, on the basis above, correspond to a duty of 772 millions of foot-pounds per 100 pounds of coal. The most economical steam-engines have been claimed to give as a maximum only about 130 millions, on the same basis, equivalent to an ultimate efficiency of ($130 \times 100 \div 772 =$) 16.84 per cent of the heat in the steam, and but ($16.84 \times .69 =$) 11.62 per cent of the calorific value of the fuel. If

$D =$ duty in foot-pounds per 100 pounds of coal,
$H =$ the height of lift per gauge,
$T =$ the initial and t the final temperatures, respectively,

* Centennial Report; Group XX; 1876.

then

$$D = \frac{1,000,000}{T - t + .0013H}.$$

The standard taken by British engineers in measuring duty is usually the number of pounds raised one foot either by 94 pounds of coal, or by one "hundredweight" (112 pounds). On this basis the duty is computed from the ascertained weight of fuel per I. H. P. per hour, thus:

$D =$ duty in millions;
$F =$ weight of fuel per I. H. P. per hour;
$D = \frac{186.12}{F}$, where the unit is 94 lbs.;
$D = \frac{221.76}{F}$, where in cwts.

156. **The Various Objects, in Detail,** sought in test-trials are determined in part by the specific purpose of the trial; but in nearly every case they require the measurement of power obtained, and of cost of obtaining it, expressed either in money or in fuel consumed; and this means the exact weighing of fuel, the measurement of water used, the determination of the quality of the fuel and of the steam made, and its quantity; and the distribution of the stored energy, by the engine, in useful and in wasteful directions. Every quantity must be exactly measured which has importance in relation to the question at issue, and the data collected must be secured in such manner that their magnitudes may be readily introduced into the computations, and may, if question arises, be readily checked and verified. The methods of determination and of record thus become matters of importance and careful study, and the application of the greatest possible ingenuity and skill are demanded in the effort to devise an acceptable and reliable system. In studying the efficiency of capital, it is first necessary to consider the elements of cost of power.

157. **The Maker and the Methods of the Trial** are usually subjects of stipulation in the contract. In some cases the

parties to the contract merely agree that any question arising shall be left to the arbitrament of a board of three competent men—one appointed by each, and the third by the first two appointees. In other cases it is agreed that a known and reputable expert shall either conduct the trial or shall direct the appointments. Most frequently, when the work is very important, the contract provides for a board of three experts, who are usually named in the instrument, and also prescribes the method of conducting the trial. Those who are chosen for so important a duty should always be engineers of known ability, integrity, energy, and experience; whose professional practice has afforded them opportunities to become familiar with the best methods, and has given them skill in their employment. Standard and well-known and universally-accepted methods should be prescribed whenever possible.

158. The Character of Report demanded of the engineers conducting the trial is determined by the purpose of their work. In every case it should be simple, easily comprehended, so far as that may be possible, by a non-expert reader; and it should give all essential data, processes, and results, in such manner that, in case they are called in question, they may be verified completely and with certainty. The matter sought to be settled should be defined with precision and clearness, and the whole statement should be as concise and as absolutely free from irrelevant matter as possible. In the endeavor to explain presumably obscure points, even, opportunity will be found for the exercise of a good judgment and great discretion. Good examples of the best forms of report will be presented later.

Recent standard methods have approximated very closely to the exactness and accuracy of the processes of the physicist or the chemist. In fact, the apparatus and processes of these investigators are now adopted by the practitioner, and the young engineer himself is now almost invariably trained by the physicist and chemist, in all the exact methods of the laboratory, in his preliminary professional studies in the technical school. The determination of temperatures and the ascertainment of weights are conducted by all the accepted methods of

exact scientific measurement, and the analysis of fuels, of gases, and of ashes is bringing into play and application some of the most interesting and accurate operations of the chemical laboratory.

159. **The Apparatus** employed in test-trials of engines and boilers should be made and standarized with all the care exacted in any other department of physical research. Instruments of every class used in such investigations should be carefully selected, and from the stock of the best makers; they should be inspected, tested, and compared with standard instruments of known excellence and accuracy, and any permanent errors in their operation and method of application should be recorded with every possible care. Scales should be compared with the legal standards; tanks and other volume-measures should themselves be very nicely measured; thermometers should be calibrated; indicators should be tested, hot as well as cold; and dynamometers should be measured up with similar accuracy.

160. **Methods of Application** of the instruments used are commonly well determined by experience and settled by custom. Such serious variations of result may be due to error in this matter that the study of the effects of differences of practice becomes an important part of the task of the engineer-expert. A fault in location of a thermometer, or in the setting up of an indicator, may produce quite sensible, and even serious, differences in the magnitude of data so obtained.

161. **The Data Needed and Computations** required in the determination of the character and performance of any engine or boiler, or of both, are obtained by means of continuous or closely successive readings from all instruments employed, so taken and recorded as to give a substantially exact record of the whole period over which it has been concluded to extend the trial. These observations must be sufficiently frequent and numerous to permit the computation of very accurate averages, and, where graphical methods are, as is now common, adopted, to permit the construction of smooth and complete curves showing the whole period of the trial. Every continuous pro-

cess should thus be capable of representation by a similarly continuous record, and in such manner that every computation required may be readily effected. The results of computation will then furnish accurate numerical measures of every quantity needed to determine whether the contract has been fully complied with.

162. **Trials to Determine Economy and Efficiency** are most common, and are generally of most importance. Few contracts for important steam apparatus are made which do not include a specification of the degree of economy demanded in the use of both steam and of fuel, and often of a system of test-trial as well. In such cases it becomes necessary for the engineers conducting the trial to ascertain with precision the quantity of fuel, of steam, or of heat-energy supplied to the engine, the amount of energy converted into the mechanical form, the proportion of heat and of mechanical power wasted, the methods and extent of waste, in full detail, and the power applied by the machine to such useful purpose as it is designed to subserve. The measure of the benefit received by the user of the engine is the useful work performed; the measure of its cost to him is what he pays in fuel, steam, or heat-energy, and the money-equivalent of this supply, and of all incidental costs, such as rent, attendance, wear and tear, insurance and taxes, and depreciation. A comparison of the mean continuous cost with the average value of power supplied for useful work exhibits the real value of the machinery to its purchaser and user. Such a trial is only complete when it determines accurately the following:

EXPENDITURES: Quantities and Costs.	RECEIPTS: Quantities and Costs.
Fuel;	Useful Work;
or	Wasted Work and Heat:
Steam;	(*a*) Friction of Engine;
or	(*b*) Heat lost externally;
Heat-energy;	(*c*) Heat lost internally and
supplied by the user.	rejected from the system.

163. Steam-boiler Efficiency is not difficult of definition when the nature of the quantity to be measured is itself first understood. There are, however, as will be presently seen, several different efficiencies of the steam-boiler, as of the steam-engine; and it is important that each be distinctly defined before a study of either, or of total efficiency, can be made. In general, it may be said that efficiency is measured by the ratio, in common or similar and definitely related terms, of a result produced to the cost of its production. As, in the study of the steam-engine, either efficiency is measured by the ratio of work done in the specified manner to the work or work-equivalent expended in doing it; so, in the case of the steam-boiler, either efficiency is measured by the ratio of a heat-effect, or its equivalent, to the quantity of heat, actual or latent, paid for its accomplishment.

In some cases it is not practicable to thus establish a numerical value of an efficiency; and it can only be shown that efficiency, in the sense of quantity of result compared with magnitude of means used, is increased or decreased by the operation of defined phenomena, or by conditions which can be specified. A common measure cannot always be found, or an exact law of relation established.

164. Trials to Ascertain Power or Maximum Capacity to do useful work are often, perhaps usually, made under the same contract as are those to determine efficiency of engine. Steam-machinery is commonly guaranteed, both as to economy and capacity. Such trials are sometimes made at the same time with efficiency-trials; but the maximum power of engines and boilers is seldom, if ever, that at which best economy is obtainable. A single trial is made when the power guaranteed is that of normal working, and that for which the guarantee of economy is made by the contract. A trial for capacity simply is one in which the power only need be measured, and its cost, unless specifically demanded, is not determined. The methods employed, so far as they go, are the same as in the preceding and more complete kind of trial.

The actual power of steam and of boilers evidently depends

upon the efficiency of the method of application, and on the apparatus employed. The quantity of heat-energy supplied to the engine and yielded by the generator has been seen to be easily calculable by simply multiplying the quantity of heat given to the steam, by the fuel, by the mechanical equivalent of heat. The amount available as energy may be the total quantity so supplied, as when the steam is condensed in heating buildings or otherwise, and is returned as feed-water to the boilers; or it may be any less amount, according as the method of utilization is more or less effective. Where no constant value can be assumed for the efficiency of the system employed, it is sometimes, nevertheless, found to be important to establish a standard conventionally. Thus, convention has established the unit horse-power of steam-boilers, in order to afford a standard of comparison in test-trials, and to give a means of rating boilers by the designer, the builder, or the purchaser and user.

The operation of boilers occurs under a wide range of actual conditions—the steam-pressure, the temperature of feed-water, the rate of combustion and of evaporization, and, in fact, every other variable condition, differing in any two trials to such an extent that direct comparison of the totals obtained, as a matter of information regarding the relative value of the boilers, or of the fuel used, becomes out of the question. It has hence gradually come to be the custom to reduce all results to the common standard of weight of water evaporated by the unit-weight of fuel, the evaporation being considered to have taken place at mean atmospheric pressure, and at the temperature due that pressure, the feed-water being also assumed to have been supplied at the same temperature. This, in technical language, is said to be the "equivalent evaporation from and at the boiling-point" (212° F., 100° Cent.). This standard has now become generally incorporated into the science and the practice of steam-engineering. The "Unit of Evaporation" is one pound of water at the boiling-point, evaporated into steam of the same temperature. This is equivalent to the utilization of 966, nearly, British thermal units per pound of

water so evaporated. The economy of the boiler may thus be expressed by the number of units of evaporation obtained per pound of combustible.

165. **The Quantities Measured and Results** sought to be secured are thus, in detail, as follows for any complete trial:

When the trial includes, as is most frequently the case, a trial of the boiler, the combined efficiency of boiler and engine being the final determination, arrangements must be made in advance to ascertain exactly the weights of fuel, gross and net, —coal and ash for example; the weight of water supplied as "feed;" the weights, temperatures, and pressures of dry steam, and weight of entrained water; the temperatures of furnace, flues, and chimney; of superheating steam, if it be so heated; the power of the engine, gross and net; the friction of engine; the wastes by cylinder-condensation and otherwise; the steam-pressure in boiler and steam-chest; and the continually varying pressures in the working cylinder throughout the whole cycle, revolution by revolution, of the engine. Each of these quantities is measured at specified intervals, and a comparison of mean values of power usefully applied, and of expenditures made to produce it, gives the measure of the economy attained.

At the same time that the gross power developed by the action of the steam in the cylinder, the indicated power, is measured, the diagrams taken furnish the means of ascertaining precisely how the pressures and volumes of the steam simultaneously vary within the engine, and thus give a clue to, and usually a fairly exact determination of, the setting and motion of the valves and the extent to which the distribution of steam is such as will best conduce to economical working. These diagrams also enable the engineer to compute with considerable accuracy the volumes and weights of the steam at any and at every point in the stroke. A comparison of the quantities so calculated with the actual measures obtained at the boiler, or before the steam enters the cylinder gives the measure of the quantity condensed in the cylinder as the piston moves forward, and of the later re-evaporation. The cylinder-wastes are thus also determinable with a fair degree

of accuracy. These "cards" also exhibit the amount of back-pressure, and this measures the resistances in the exhaust-passages and at the condenser, if there be one, and thus afford a means of criticism of the design and construction of the engine in this respect. Similarly, the difference between the steam-pressures in the cylinder and in the steam-chest and the exhaust-chest is a measure of the losses in the steam-passages.

166. **General Schemes of Trial or Tests of Engines** are adopted by engineers which, while varying in detail, all closely resemble each other in their main purposes, and are somewhat similar in methods. They commonly include boiler-trials as the only practicable and satisfactory means of ascertaining the quantity and quality of steam supplied, and the cost of power in steam, fuel, and money. They invariably involve the application of the indicator or the dynamometer, and, if complete, of both, for power measurements. When the question to be solved is simply the efficiency of engine, or of engine considered dynamically, of the engine as a machine, a comparison of the indicated with the dynamometric power gives the solution; but when the thermal efficiency and the efficiency in transformation of energy is to be measured, the measurement of the quantity of energy supplied in the form of heat, and hence a boiler-trial, must necessarily form a part of the operation. All general systems may therefore be said to involve the whole series of determinations of quantity already indicated; but the details have not yet been authoritatively prescribed in such manner as to fix a standard system or standard methods. The example of the most experienced and distinguished practitioners is, however, gradually producing a tolerably well settled custom in the more important of the several operations involved. Some such methods and some general plans of test-trials will be later described.

167. **Steam-boiler Trials,** apart from the engine-test, and made for the purpose of ascertaining the quantity and quality of steam made, its cost in fuel and in combustible matter contained therein, and the efficiency of the boiler and of its heating-surface, are now very generally made by a fairly well-

recognized system. In the United States and in Germany, particularly, such methods are now made to follow very generally the prescribed order of procedure devised and published by engineers of recognized standing. Such a standard system is that proposed by the Committee of the American Society of Mechanical Engineers, and this standard will be that accepted in this work.*

168. Engine Trials may or may not include determination of boiler performance and efficiency; but if they are to be satisfactorily complete, measurements of the quantity and quality of the steam supplied are as essential as any other determinations of quantity. In some cases only a comparison of the work done, with its cost in fuel, is called for; but in this case the total efficiency so obtained cannot be analyzed into the two factors, engine efficiency and boiler efficiency, and it is impossible to say to what extent engine or boiler is responsible for the final results obtained. The complete investigation of the action and performance of the engine, as a heat-engine and prime motor, must always include some method of obtaining a measure of the amount of heat-energy supplied to the machine; the proportion of that energy which reaches the engine in available form; the distribution and disposition of the available part; the extent to which it is converted into useful work and into wasted power; the amount in detail of the various wastes; the method as well as extent of wastes; and the ameliorating or exaggerating effect of any observable accidental or purposely produced variations of condition and of operation upon the wastes, the economies, and the several efficiencies of the engine.

It is thus important that ways should be found and methods practised that will determine the quantity and quality —whether wet or dry—of the steam supplied, the pressures and volumes of every stage of transfer and of transformation,

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* Transactions American Society of Mechanical Engineers, vol. VII, 1884. A Manual of Steam-boilers, by R, H. Thurston (N. Y., J. Wiley & Sons) chap. XIV, pp. 484–537; Engine and Boiler Trials, 1889.

and the quantities of heat stored, conveyed, and utilized or wasted.

169. **Engine and Boiler Tests** are thus necessarily commonly conducted simultaneously in the settlement of important contracts, and essential data can only be thus secured. Where the quantities to be ascertained and measured are not likely to vary greatly with period of operation, a trial of a few hours' duration will answer all purposes. Gas-engines are often tested a single hour, and five hours is quite as long as is often desirable. Steam-engine and boiler trials are seldom less than ten hours in length, often occupy a full day of twenty-four hours, sometimes last a week, unintermittedly, night and day, and it is even sometimes prescribed that the more important data shall be recorded for several months or for a year at a time. Ordinarily, a ten-hour trial is quite sufficient, if properly conducted.

170. **The Apparatus of Steam-boiler Trials** consists of tanks to receive and in which to weigh the feed-water; scales with which to effect these measurements and to weigh fuel and ashes: thermometers with which to determine the temperature of the water and steam, and pyrometers for the furnace-flue and chimney temperatures; and it is now usual to employ a calorimeter with which to determine the condition of the steam, and to measure the proportion of entrained water. Before the systems of boiler trial usually adopted are employed, it will be necessary to understand the methods of use and of calibration and of standardizing these various kinds of apparatus, the sources from which they may be obtained, and the best methods of their application to the securing of the needed data. It is usually thought best to *weigh* all water, rather than measure its volume. If measured, it should be carefully noted that its variation of density with temperature is considerable, and sufficient to introduce observable errors if a constant density is assumed.

The Apparatus of Engine-testing consists of steam-engine indicators, dynamometers, counters and gauges, and good timing instruments. The use of these instruments and the

methods of test and correction are simple; and, with the exception of the indicator, none demands very extended notice. The indicator, however, is an instrument which must be made with the utmost possible care and skill, and the study and interpretation of its diagrams is a matter demanding some skill, knowledge, and experience.

Uniformity of operation and maximum efficiency are best attainable during a trial when a system of record is adopted which allows of that regularity being shown at all times; and records in proper form are the best possible security against error of observation. Graphical methods should be adopted wherever practicable. Such methods of record exhibit most satisfactorily the accordance with or the deviation from the uniformity of operation considered so desirable on the score of efficiency and accuracy.

171. **Standard Test-trials** are made under established systems, and in accordance with codes of regulations which are accepted as representing a satisfactory system of procedure. In such cases the first step is to settle upon a standard of measurement and comparison that may be accepted by all who may be interested in the result.

The Committee of Judges of the Centennial Exhibition, to whom the trials of competing boilers at that exhibition were intrusted, adopted the unit of power 30 *pounds of water evaporated into dry steam per hour from feed-water at* 100° *Fahrenheit, and under a pressure of* 70 *pounds per square inch above the atmosphere*, these conditions being considered to represent fairly average practice. The now-accepted unit of boiler-power, in the code constructed for the American Society of Mechanical Engineers,* is the equivalent of the Centennial Standard, and in all standard trials the commercial horse-power is taken as *an evaporation of* 30 *pounds of water per hour from a feed-water temperature of* 100° *Fahrenheit into steam at* 70 *pounds gauge-pressure*, which is equal to 34½ "units of evaporation," that is, to 34½ pounds of water evaporated from a feed-

* Transactions; vol. VI, 1884.

water temperature of 212° Fahr. into steam at the same temperature. This standard is taken as the equivalent of 33,305 thermal units per hour.*

172. Instructions and Rules governing the standard system of boiler-trial, prepared by a committee of the American Society of Mechanical Engineers, may be taken as a good illustration of such regulations as, in one form or another, have become generally accepted, and such trials are commonly made in the manner thus prescribed.†

173. Precautions are to be taken in every possible way to prevent and avoid irregularities in the conduct of the trial and errors of observation.‡

In preparing for and conducting trials of steam-boilers the specific object of the proposed trial should be clearly defined and steadily kept in view; and as suggested by Mr. Hoadley—

(1) If it be to determine the efficiency of a given style of boiler or of boiler-setting under normal conditions, the boiler brickwork, grates, dampers, flues, pipes, in short, the whole apparatus, should be carefully examined and accurately described, and any variation from a normal condition should be remedied, if possible, and if irremediable, clearly described and pointed out.

(2) If it be to ascertain the condition of a given boiler or set of boilers with a view to the improvement of whatever may be faulty, the conditions actually existing should be accurately observed and clearly described.

(3) If the object be to determine the relative value of two or more kinds of coal, or the actual value of any kind, exact equality of conditions should be maintained if possible, or, where that is not practicable, all variations should be duly allowed for.

* An evaporation of 30 pounds of water from 100° F. into steam at 70 pounds pressure is equal to an evaporation of 34.488 pounds from and at 212°; and an evaporation of 34½ pounds from and at 212° F. is equal to 30.010 pounds from 100° F., into steam at 70 pounds pressure.

† See the Author's Manual of Steam-boilers, p. 493, § 256; and Engine and Boiler Trials, p. 46, § 33.

‡ The appendix to the report above quoted should be read in this connection.

(4) Only one variable should be allowed to enter into the problem; or, since the entire exclusion of disturbing variations cannot usually be effected, they should be kept as closely as possible within narrow limits, and allowed for with all practicable accuracy.

Blanks should be provided in advance, in which to enter all data observed during the test. The preceding instructions contain the form used in presenting the general results. Records should be, as far as possible, made in a standard form, in order that all may be comparable.

Several methods of weighing fuel have been found very satisfactory, but it should be an essential feature that the weights shall be made by one observer and checked by another, at as distant a point as is convenient. The weighing of the fuel by one observer at the point of storage, and the record at that point of times of delivery, as well as of weights of each lot, and the tallying of the number and record of the time of receipt at the furnace-door, will be usually found a safe system. The failure to record any one weight leads to similar error, and can only be certainly prevented by an effective method of double observation and check.

The same remarks apply, to a considerable extent, to the weighing of the water fed to the boiler. A careful arrangement of weighing apparatus, a double set of observations, where possible, and thus safe checks on the figures obtained, are essential to certainty of results. With good observers at the tank, and with small demand for water, a single tank can be used; but two are preferable in all cases, and three should be used if the work demands very large amounts of feed-water, as at trials of very large boilers, or of "batteries." The more uniform the water-supply, as well as the more steady the firing, the less the liability to mistake in making the record.

The blanks, which will be found in the Appendix to Part I, were prepared by the Author for use in laboratory as well as professional work. That in the Appendix to this volume is a blank designed for condensed reports on tests made in the Sibley College laboratory.

174. The Results of Trials actually conducted under acceptable conditions, and with all the precautions which have been suggested, are illustrated by the following examples:

The first case is a trial which was carried out in accordance with the above programme. The measurements of the feed-water were made by passing the water through a Worthington metre into two wooden tanks located on Fairbanks' Standard Platform Scales. The pipe connections were so arranged that one tank could be filled and weighed while the other tank was being emptied into the boiler.

Each tank was filled once every half-hour. As soon as the tank was full and the pumping into the boiler commenced, the temperature of the feed-water was taken by sensitive thermometers reading to one tenth of a degree.

All water measurements, as in all instances of careful work, were made by weight rather than by volume, and systems of checking were devised and practised whenever practicable. The apparatus was all carefully standardized, and repeatedly re-examined and tested as opportunity offered.

The measurements of the coal were effected by weighing the coal previous to its being wheeled into a pile in the coal-room. The second weighing was made when the coal was fed into the furnace. As far as it was possible, the furnace was supplied with coal at intervals of every half-hour, so as to correspond as nearly as could be to the feeding of the water.

After the completion of the test, a careful analysis of the coal was made, to determine upon a sufficiently large scale its calorific power and the quantity of contained moisture. The steam from the boiler was condensed by means of a continuously acting calorimeter, formed by placing four tanks on Fairbanks' Standard Platform Scales. The steam from the boiler was passed through a surface-condenser having a condensing surface of 631 sq. ft. As fast as the steam was condensed from the boiler it was received in small tanks located on platform-scales. These tanks were similar in size to the feed-water tanks, and were so arranged as to be filled and

emptied once every half-hour, one tank receiving the condensed water from the boiler while the other was being emptied.

The condenser was supplied with a large volume of cold water from a weir just outside of the works, and after flowing through the condenser, and thereby cooling the steam and receiving therefrom the contained heat, this water was caught in two large tanks placed on platform-scales. These tanks were also arranged so that one tank could be emptied while the other was being filled, and were of sufficient capacity so as to insure catching all of the water required for half an hour's run in the condenser. The temperature of the inlet water of the condenser, of the outlet water, and of the condensed steam were carefully noted by means of thermometers reading to a tenth of a degree. Readings of the inlet water and of the condensed steam were taken once every half-hour at the same time that the quantities of the water in the tanks were weighed. Inasmuch as the outlet to the condenser varied considerably in temperature, readings on this were taken every five minutes during the entire time of the test. It will thus be seen that a very correct average of the amount of heat given to the condenser was obtained. The quantity of air supplied by the blowers to the furnace was measured by continuously acting anemometers placed in the supply-pipes. The readings of the anemometers were checked by means of the number of revolutions of the blowers and their cubic feet per revolution.

The steam-pressure was kept by a recording pressure-gauge, which was checked by an exceedingly delicate and sensitive gauge, which previously, and subsequently to the test, was carefully verified by means of a mercury column. Constant records of the hygrometer, barometer, and thermometers, both in the boiler-room and of the external air, were kept during the entire period of the test.

It will be seen from the above that all of the processes and measurements were kept in duplicate in such a way as to afford a constant check on each other, and preclude the possibility of any errors.

The quality of the steam was determined by a calorimeter.

The following is a brief condensed summary of results:

DATA AND EFFICIENCY.

Total heat of boiler	64,536,613	heat-units.	
Steam	42,933,141	" "	66.6 per cent.
Heat escaping in flue-gases	9,669,036	" "	15 " "
Radiated heat	5,162,939	" "	8 " "
Heat to vaporize moisture in coal	141,372	" "	0.2 " "
Heat to vaporize moisture in air supplied to furnace	345,978	" "	0.4 " "
Leakage	3,531,645	" "	4.0 " "
" from pump	127,936	" "	0.2 " "
Heat absorbed by fire-brick	2,581,645	" "	4.0 " "
Unaccounted for	1,092,941	" "	1.6 " "

In the trial of an upright boiler reported on by Sir Frederick Bramwell, in 1876, coke being used as the fuel and wood in starting the fires, the following data* were obtained:

Ash and moisture	43.79 lbs.
Combustible	194.46 "
Total fuel	238.25 "
Air used per pound combustible	17½ "
Heat generated, net	2,798,312 B. T. U.
" " per lb. fuel	11,745 " "
" " available, net	2,101,700 " "
Water evaporated	1,620 lbs.
The efficiency of the furnace was	0.643 "

The balance-sheet stands thus:

Per Cent.		
	Dr.	
	Available heat	2,101,700 B. T. U.
	Cr.	
88.29	Heat expended in exaporation,	1,855,900 B. T. U.
7.03	Displacing atmosphere	147,720 " "
3.35	Loss by conduction and radiation	70,430 " "
.05	Heat in ashes	1.129 " "
1.26	Unaccounted for	25,521 " "
100.00		2,101,700

* Conversion of Heat into Work. Anderson.

175. The Quality of Steam made in any boiler, or as supplied to an engine, is hardly less important than the quantity. When the steam is required for heating purposes simply, or when all the heat issuing as waste, necessary or other, from the exhaust-ports of a non-condensing engine-cylinder can be utilized for useful and paying purposes, this is a matter of

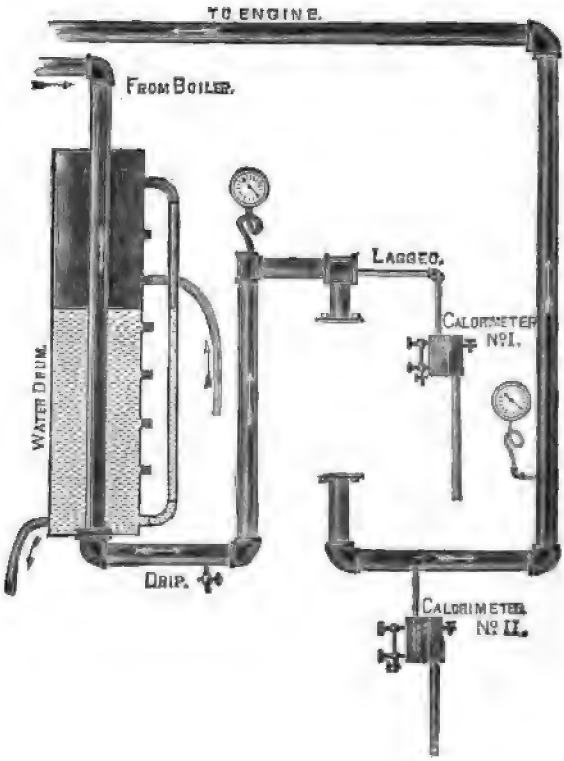

FIG. 189.—TRIALS OF SEPARATORS.

no importance ; but when it is essential that the engine shall have maximum efficiency, the quality of the steam becomes exceedingly important. The determination of the quality of steam by any boiler is thus as important as the measure of its apparent evaporation.

Methods and result of experimental investigations of the

efficiency of separators in insuring dry steam are illustrated by the following: *

To obtain the needed range of quality of steam, the experimenters, Messrs. Brill and Meeker, enclosed a vertical section of pipe with a drum or cylinder. As may be seen from the sketch, this drum had several openings along the side to permit water being introduced at various heights, and an outlet was arranged at the bottom, thus maintaining a good circulation.

It was concluded to make the runs of twenty minutes' duration, repeating as often as was found desirable. The gauges and thermometers were read simultaneously at intervals of five minutes. The gauges were calibrated occasionally, in order that no change could occur unnoticed.

The volume of each of the separators was determined by filling with water, expecting to find that to a certain extent, at least, the efficiency of separation depended upon the volume; but no such relation was discovered, owing to the differenees of form.

The separators are designed by the letters *A*, *B*, *C*, *D*, *E*, and *F*. The qualities of the steam before and after separation and the efficiency, taken as the ratio of per cent of water removed to per cent of water in the entering steam, are given in the table and the diagrams on p. 606:

The separators rank, in order of efficiency for quality of steam above ninety per cent, as follows: *B*, *A*, *E*, *D*, *C*, and *F*; but with more than ten per cent of moisture, *E* shows a rapid decrease in efficiency, soon losing its place in the list. The curves of the four separators *B*, *A*, *D*, and *C* have the same general form; each separator reaching a maximum efficiency at about 35 per cent of moisture.

No marked decrease in pressure was shown by any of the separators, the most being 1.7 pounds in *E* and ranging from this to .6 pound in *B*.

This investigation clearly shows that although changed

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* Power; Prof. R. C. Carpenter; July, 1891; p. 9.

EFFICIENCY OF SEPARATORS.

B

Quality Before	Quality After	Eff. per cent
97.5	99.0	60.0
96.4	99.0	72.2
93.6	99.0	84.4
93.4	99.0	84.8
87.0	98.8	90.8
83.3	98.4	90.4
78.1	98.8	94.5
76.9	98.6	93.9
76.0	98.8	95.0
72.7	97.8	91.9
69.4	99.0	96.7
66.1	98.8	96.4

A

Quality Before	Quality After	Eff. per cent.
98.0	98.0	0.
97 8	99.1	59.1
96.1	98.4	59.0
95.3	98.2	61.7
90.1	98.0	80.0
80.4	98.1	90.3
79.5	98.2	91.2
63.0	98.0	94.6
58.0	98.0	95.2
54.4	98.1	95.8
54.3	97.9	95.4
51.9	98.4	94.6

E

Quality Before	Quality After	Eff. per cent.
98.1	98.5	21.1
97.8	98.3	22.7
94.8	98.1	63.5
93.6	98.1	70.3
88.4	90.2	15.5
82.4	90.4	45.5
74.6	81.6	27.5
73.2	79.2	22.4
71.8	84.9	46.4
68.6	79.3	34.1

D

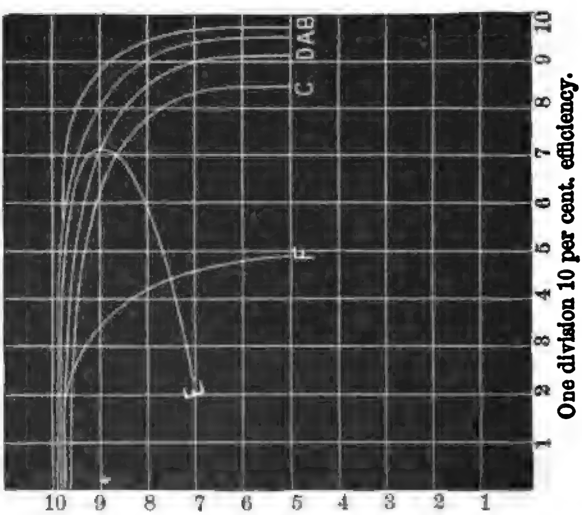
Quality Before	Quality After	Eff. per cent.
96.1	97.4	33.8
93.7	97.2	55.5
89.6	95.8	59.6
88.9	96.4	67.6
84.3	96.4	77.1
83.0	96.8	81.2
81.7	97.9	83.5
75.9	95.5	81.3
74.6	98.2	92.9
72.2	95.8	84.9

C

Quality Before	Quality After	Eff. per cent.
95.6	95.8	4.5
96.8	98.4	50.0
91.9	98.6	82.7
90.6	93.7	33.0
88.9	95.7	61.2
88.5	94.5	68.0
87.7	95.1	60.2
78.2	95.6	79.8
78.2	94.7	75.7
74.4	94.8	79.7
73.2	95.3	82.4
67.1	94.8	84.2

F

Quality Before	Quality After	Eff. per cent.
97.7	97.9	8.7
97.5	97.6	.4
94.7	96.0	24.5
88.9	92.1	28.8
82.9	89.2	36.8
81.6	84.3	15.8
80.9	89.2	43.4
79.3	87.2	38.1
75.8	85.9	41.7
70.4	84.1	46 2



One division 10 per cent. quality.

FIG. 190.—EFFICIENCY OF SEPARATORS.

direction, reduced velocity, and perhaps centrifugal force are necessary for good separation, some means must also be provided to lead the water out of the current of the steam.

If such provision is not made, momentary separation may occur, but the water will be again taken up by the rapidly moving steam. The high efficiency obtained from *B* and *A* was considered to be largely due to this feature. In *B* the interior surfaces so formed as to catch the water thrown out of the steam and lead it to the bottom. In *A* the water comes in contact with a perforated diaphragm, through which it passes into a detached space below. In *D* this is accomplished by means of a >-shaped diaphragm, which throws the water back into the corners out of the current of steam. *E* depends for its action upon a pendant diaphragm placed across a greatly enlarged section of a horizontal pipe. In *F* separation is intended to be accomplished by centrifugal action.

As a result of the investigation it is concluded that steam separators may be constructed which will furnish a uniformly good quality through as great a variation in quality as will be found in practice.

An excess over 5 per cent moisture, a "quality" below 95 per cent, is rarely allowed in good steam-engine practice.

Trials not including calorimetric measurement of the water entrained with the steam should be rejected in any important case. Reports of extraordinary economy are often based on this kind of error. The experiments of M. Hirn at Mulhouse showed an average of about 5 per cent priming; Zeuner makes it approximately from $7\frac{1}{2}$ to 15 per cent; while the experiments of the Author* at the American Institute in 1871 gave from 3 to 6.9 per cent.

A recently devised method of measuring the amount of moisture in the steam is to introduce into the boiler with the feed-water sulphate of soda, and at intervals to draw from the lower gauge-cock a small amount of water, and also from the steam, condensing either by a coil of pipe in water or a small

* Trans. Am. Inst., 1871; Journal of the Franklin Institute, Oct. 1871; Vienna Reports, 1875; vol. III.

pipe in air. A chemical analysis gives the proportion of sulphate of soda in each portion, and the quotient of the proportion of sulphate of soda in the portion from the steam by the proportion in that from the water gives the ratio of water entrained, as steam does not carry sulphate of soda, which is only brought over by the hot water entrained. This method was used by Professor Stahlschmidt at the Düsseldorf Exhibition Trials.

176. **The Calorimeters** used in determining the quantity of moisture in steam have several forms. The objects sought to be attained in their construction are: The exact measurement of the weight of steam received by them from the boiler, and of its temperature and pressure at the boiler; the determination of the weight of water used in its condensation, and the range of temperature through which it is raised in the operation; the reduction of wastes of heat in the calorimeter to a minimum, and the exact measurement of that waste if it is sensibly or practically noticeable.

The Barrel or Tank Calorimeter, as employed by the Author, is the simplest form of this instrument which has been produced. It consists of a strong barrel or tank, of hard wood, absorbing little of either water or heat, and having a movable cover. This tank is mounted on platform-scales capable of accurate adjustment and having as fine readings as possible. It is filled with water to within about one fourth its height from the top, and the steam is led into it through a rubber tube or hose of sufficient capacity to supply the steam to the amount of one eighth or one tenth the weight of the water in three or five minutes. A steam-gauge of known accuracy gives the boiler-pressure, and the corresponding temperature and total heat of the steam are ascertained from the steam-tables. In the original form of this instrument, as constructed and used by Hirn, a stirring apparatus was employed; but the steam may be so introduced, as above, as to render this accessory unnecessary, producing a rapid and general circulation.

177. **The Theory of the Calorimeter** is as follows:

Each pound of saturated steam transferred to the condens-

ing-water the quantity of heat which had been required to raise it from the temperature of the water of condensation to that due to the pressure at which it left the boiler, *plus* the heat required to evaporate it at that temperature. Each pound of water gives up only the quantity of heat required to raise it from the temperature of the water of condensation to that of the steam, with which it is mingled. The total amount of heat is made up of two quantities, therefore, and a very simple algebraic equation may be constructed, which shall express the conditions of the problem:

Let H = heat-units transferred per pound of steam;
h = heat-units transferred per pound of water;
U = total quantity of heat transferred to condenser;
w = total weight of steam and water, or of feed-water;
x = total weight of steam;
$w - x$ = total weight of water primed.

Then

$$Hx + h(w - x) = U; \quad \text{or} \quad x = \frac{\frac{U}{h} - w}{\frac{H}{h} - 1} = \frac{U - wh}{H - h}.$$

Substituting the proper values in this equation, we determine the absolute weights and percentages of steam and water delivered by the boiler.

Or, let

Q = quality of the steam, dry saturated steam being unity;
H' = total heat of steam at observed pressure;
T = " " " water " " "
h' = " " " condensing water, original;
h_1 = " " " " " final.

And we have the equivalent expression, as written by Mr. Kent,

$$Q = \frac{1}{H' - T}\left[\left(\frac{W}{w}(h' - h_1\right) - \left(T - h_1\right)\right].$$

The value of the quantity U is obtained by multiplying the weight of water in the calorimeter originally by the range of temperature caused by the introduction of the steam from the boiler. Mr. Emery employs another form, as below, in which Q is the quality of steam as before; W the weight of condensing-water; w the weight added from the boiler; T the temperature due the steam-pressure in the boiler; t the initial and t_1 the final temperature of the calorimeter; l the latent heat of evaporation of the boiler-steam; and x the weight of steam corresponding to l. Thus

$$x = \frac{W(t_1 - t) - w(T - t_1)}{l}; \quad y = 100\frac{w - x}{w};$$

and

$$Q = \frac{x}{w} = \frac{W(t_1 - t) - w(T - t_1)}{lw}.$$

The heat in a pound of steam can also be determined by the thermometer, provided the steam has a higher temperature than that due to its pressure, i.e., is superheated. If the specific heat of steam be assumed 0.48, and the reading of the thermometer T', the degree of superheat is $T' - T$, which, reduced to B. T. U., equals $0.48(T' - T)$, so that the total heat-units contained in one pound of the steam is equal to

$$H + 0.48(T' - T).$$

Two calorimeters have been devised on this principle: the superheating calorimeter of Barrus, in which the heat is supplied by extraneous means, and the throttling calorimeters of Peabody and Barrus, in which the heat required for superheating is obtained by reducing the pressure, which, being accompanied by a corresponding reduction of boiling-point, liberates heat sufficient to evaporate a small percentage of moisture. In the *Superheating Calorimeter:* In this, if $t_4 - t_1$ is the gain of temperature in the live-steam pipe, and $t_2 - t_3$ is the loss of

temperature by the superheated steam, we have, neglecting radiation,

$$1 - x = .48[t_2 - t_3 - (t_4 - t_1)] \div l.$$

In the *Throttling Calorimeter*, where the steam is superheated by expanding, we have, calling the sensible heat s,

$$x = [H + .48(t' - t) - s] \div l.$$

In both the above, the effect of an error of one degree in temperature is to make an error in x of .06 of one per cent, while an error of 9 degrees in temperature will affect the value of x but $\frac{1}{2}$ per cent. The boiling-point should be correctly determined, however, especially if the amount of superheating is small.

The effect of an error in gauge-reading of one pound will make one half the error as with the other class of calorimeters.

Conditions for determination of the moisture to within one half of one per cent require:

(1) Scales that weigh accurately to half of one per cent of the quantity to be weighed.

(2) Thermometers that give accurate determinations to one tenth degree F.

(3) An accurate pressure-gauge.

(4) Correct observations of the resulting quantities.

(5) Determination of radiation-loss from the calorimeter.

A *Conventional Unit* is proposed by Hallauer, by which to measure the weight of dry, saturated, steam equivalent, under specified standard conditions, to the weight actually used in operation of the engines compared, and thus to reduce to a determinate standard the performance of all, however widely their conditions of real working may actually be. This is the weight of such steam which would demand an equal quantity of heat in its evaporation "from and at" the boiling-point under atmospheric pressure; i.e.,

$$W = \frac{Q}{l},$$

where W is the required equivalent weight, Q the quantity of heat actually supplied in the unit of time or per stroke, and l is the latent heat of vaporization at 0° C., 32° Fahr.

178. The "Coil Calorimeter" has been devised to secure more exact results in the weighing of the water of condensation than can be obtained when it is weighed as part of the larger mass. In this instrument a coil of pipe is introduced into the tank and serves as a surface-condenser in which the boiler-steam is received and condensed, and from which it is transferred to another vessel in which it is weighed by itself with scales constructed to weigh such small weights with accuracy; or the coil is removed and weighed with the contained water. In the former case drops of water may adhere to the internal surfaces of the coil and escape measurement; in the latter, the weight to be determined is increased by the known weight of the coil, and less delicacy of weighing becomes possible.

The late Mr. J. C. Hoadley constructed exceedingly accurate apparatus of the "coil" type, and obtained excellent results.

It is evident that this calorimeter may be used continuously, if desired, instead of intermittently. In this case a continuous flow of condensing-water into and out of the barrel must be established, and the temperature of inflow and outflow and of the condensed steam read at short intervals of time.

179. The Continuous Calorimeter is an instrument in which the operations of transfer of steam to the instrument and its examination are not intermitted, as is necessarily the case in the more commonly employed forms of the apparatus. The instrument being thus kept in use continuously, every variation in the quality of steam can be observed, and the number of observations can be increased to any desired extent, and, the apparatus being accurate, any degree of exactness of mean results can be attained.

One of the earliest forms of this instrument was a coil calorimeter devised by Mr. John D. Van Buren, then of the U. S. N. Engineers, and a Professor in Engineering at the Naval Academy, about 1867.

The continuous calorimeters are usually of the surface-con-

denser pattern. A *jet-condenser* calorimeter has been used by Professor Carpenter in the Sibley College laboratory with good results. The condenser was constructed of a small injector, and this instrument is found to be the best possible apparatus to insure complete and uniform intermixture of the fluids.*

The Barrus continuous calorimeter consists of a tub, in which is placed a coil of pipe; this is connected below with a drum carrying a water-gauge glass; below this is a cock connecting with a cooling coil in a tank of water. The steam is received in the coil, and condensed by the cold water flowing through the tub. The water of condensation falls into the pipe with the water-gauge glass, whence it can be drawn off as necessary. It is still farther cooled in the tank of water, and drawn off and weighed when convenient. The condensing water flows past a thermometer, in a thermometer cup, as it enters the tub, and again as it leaves; the difference in the readings of these two thermometers will be the gain $t' - t$ of the condensing water. This condensing water is caught and weighed, and gives the value of W. This system has an advantage over the barrel calorimeter in keeping the condensed steam separate from the condensing water, and thus reducing the liability of error in weighing.

The coil-continuous calorimeter, as designed by Carpenter, is a modification of the Hoadley calorimeter. It consists of a condensing coil inclosed within a casing; the condensed steam is drawn off below, and by means of a water-gauge glass can be kept at a constant level; this is finally passed through a cooling pipe. The condensing water enters at the bottom of the condenser and is discharged at the top; its temperature is taken at both entrance and exit. The radiation-loss of the condenser may be neglected if the temperature of the entering water be as much below as that of the discharge water is above the temperature of the room. Radiation-losses may usually be neglected.

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* See papers by Prof. R. C. Carpenter, Trans. Am. Society Mech. Engrs., No. CCCL, 1891; University of Mich. Technic., 1891.

In the *Superheating Calorimeter* the steam-pipe leading from the main is bifurcated, one branch passing over the flames of a large Bunsen burner, the other passing upward and finally downward, when it is jacketed by the enlargement of the first branch. The branches discharge separately, each through equal orifices, about one-eighth inch in diameter.

The theory of the calorimeter is as follows:

(1) Equal weights of steam flow through either branch of the pipe.

(2) The steam, superheated by the gas flame, is used as a jacket for the other branch, and parts with as much heat, radiation neglected, as the other gains.

(3) This amount may be measured provided the discharged steam from the central tube is superheated.

To measure this gain or loss of heat, thermometers are placed to take the temperature of steam as it enters and leaves the jacket, and on central pipe near same places.

Let $(1 - x)$ be the amount of water to be evaporated; in so doing it will take up from the jacket-steam $l(1 - x)$ heat-units. Let T be the normal temperature of the steam at the gauge-pressure; T_1 be the temperature of the superheated jacket-steam at entering, and T_2 as it leaves; let T_3 be the temperature of the superheated steam from the main pipe, and let radiation loss be r expressed in degrees F. Then if specific-heat steam be 0.48, since gain and loss of heat are equal, we have

$$.48(T_1 - T_2 - r) = l(1 - x) + .48(T - T_3);$$
$$\therefore \quad 1 - x = .48[T_1 - T_2 - r - (T - T_3)] \div l,$$

from which x may be found.

To find r, the radiation loss, shut off steam in the branch leading to the centre steam-pipe, and find reading of thermometers T_1 and T_2; after a run of equal length take $r = T_1 - T_2$. This instrument, although complicated and difficult to work, will give results accurate to one-fourth per cent in the quality.

The Throttling Calorimeter designed by Prof. C. H. Peabody, and the equations for its use, were given by their author

in 1888.* As originally designed, it consisted of a cylindrical chamber four inches in diameter, six to eight inches long, connected to the steam-pipe with a stop-valve interposed. There was also a valve in the exhaust-pipe. It was provided with a pressure-gauge and thermometers. In its operation the steam-pressure in the calorimeter was maintained at some point lower than that in the main by throttling, when, if the proportion of moisture in the steam was not great, the thermometer in the calorimeter would show superheating. If this proportion was excessive, it could not be used.

Later a standard orifice was used, and a separating-chamber was added when the range of the instrument was insufficient without it.

The equation for its use is as follows: The heat in one pound of steam before passing into the calorimeter is that due to the steam-pressure, and is $xl + s$; this is equal to that of one pound of steam at the pressure due to calorimeter-pressure plus .48 of the amount of superheat, or

$$xl + s = H_c + .48(T' - T_c),$$

in which l equals latent heat and s equals sensible heat, due to pressure in main pipe to be found in steam-tables.

H_c = total heat in one pound of dry steam at calorimeter-pressure, T' = reading of thermometer in calorimeter, and T_c = normal temperature of steam in calorimeter due to calorimeter-pressure. Care must be taken that both H and s are reckoned from 32°.

In ordinary use it delivers steam at atmospheric pressure. so that $T_c = 212°$ and H_c is 1146.6, in which case

$$x = \frac{1146.6 + .48(T' - 212) - s}{l}$$

$$= \frac{1146.6 - s}{l} + \frac{.48(T' - 212)}{l}.$$

When the steam-pressure is constant, and the degree of superheat and quality of steam alone vary, l and s will be

* Trans. Am. Society Mech. Engrs., vol. IX.

constant, and we shall have the equation of a right line in which $\frac{1146.6 - s}{l}$ is the distance above the origin that the line cuts the axis of ordinates, and $.48 \div l$ is the tangent of the angle that the line makes with the axis of abscissas. Drawing lines corresponding to the different gauge or absolute pressures, a chart may be formed very convenient for obtaining values of x without calculation. A chart formed in a similar manner, with degrees of superheat in the calorimeter as abscissa and absolute steam-pressure as ordinates, gives a series of moisture curves, and forms a chart more convenient for calculation than the first described.*

The amount of steam that will flow through the orifice at each pressure should be known; it may be weighed by condensing the steam, or computed by Napier's formula: flow, in pounds per second, $W = pa \div 70$, in which p = absolute pressure and a = area in square inches of orifice.

The limiting amount of moisture determined by the throttling calorimeter is given as follows by Peabody:

Pressure.		Priming. Per cent.
Absolute.	Gauge.	
300	285.3	07.7
250	235.3	7.0
200	185.3	6.1
175	160.3	5.8
150	135.3	5.2
125	110.3	4.6
100	85.3	4.0
75	60.3	3.2
50	35.3	2.3

The loss by radiation may be determined by attaching the calorimeter to steam known to be dry and saturated, or to steam superheated slightly, and of which the pressure and temperature, and corresponding quality, are known. The loss

* Carpenter on Calorimeters.

of temperature so determined may be used as a constant correction.

An extended use of this instrument in the Sibley College laboratories and elsewhere indicates that for a quality of steam for which it is adapted no instrument yet devised can equal it in accuracy or convenience. It is portable; its total weight need not exceed two pounds, and its extreme dimensions may be less than one foot; it is easily constructed, and its cost is insignificant. A thermometer graduated to degrees will give determinations correct to one tenth of one per cent, and no weighings are required and no corrections for weight made. The effect of errors in pressure of one pound is about equal that of two degrees on the thermometer, and has one half the effect observed in condensing calorimeters. A single careful observation, taken with calibrated instruments, is not subject to a probable error exceeding one tenth of one per cent, provided the radiation-loss be determined and corrected for.*

It is also found that the introduction of a separating-chamber in advance of the calorimeter-chamber may be relied upon, if properly constructed, to produce sensibly dry steam, and the use of a throttling-calorimeter with its thermometer becomes entirely unnecessary. Thus, if the discharged steam be condensed, both a thermometer and steam-gauge may be dispensed with. The loss by radiation and condensation must be determined, and corrected for.

Let this radiation-loss for a time equal to the duration of the run be u pounds. The weight of separated water is W; the weight of steam discharged is w. Then

$$1 - x = \frac{W - u}{W + w} \quad \text{or} \quad x = \frac{w + u}{W + w}.$$

When necessary to rely on computation, the weight of steam may be computed by Napier's rule already given, $w = \frac{1}{70} pa$ in pounds per second.

One of the simplest of the throttling-calorimeters is Heis-

* Ibid.

ler's which is illustrated in the engraving. Here the steam enters at the right, passes through a small orifice in a washer of non-conducting substance, as hard rubber, set in the "union" between the steam-valve and the instrument; drops to a low pressure and issues at the left, passing through an S-shaped chamber as seen. Its temperature in the chamber is shown by the thermometer, and its pressure by the manometer at the left. If dry it is superheated an easily computed

FIG. 191.—THE HEISLER CALORIMETER.

amount, retaining the total heat of the steam at the higher pressure. In so far as it falls short of that degree of superheating, the vaporization of a readily determined proportion of initial moisture is indicated. If not completely dried by this "wire-drawing," the steam is too wet for the use of the

instrument; but Mr. Barrus has shown that it is easy to secure accurate measurement by introducing a separating-chamber beside and in advance of the calorimeter proper.

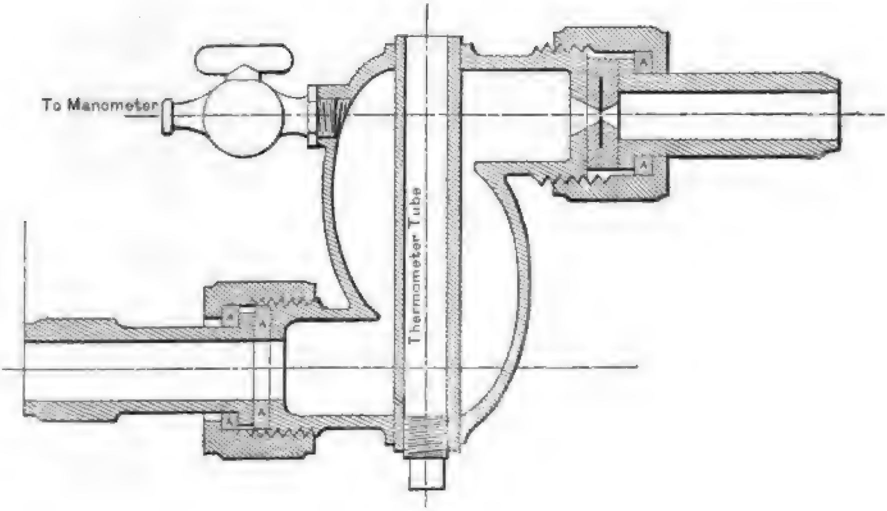

FIG. 192.—CALORIMETER.

In using these instruments, the following nomenclature, formulas, and arrangement of data may be used:

GENERAL CASE.

Quality of steam $x = \dfrac{\frac{W}{w}(t - t') - (T - t)}{l}$. Degrees superheat $S = \dfrac{x - 1}{0.48}$.

		P	T	G	W		w	t'	t	T	l	$\frac{W}{w}$	$t-t'$	$T-t$		$1-x$	x	S
Number.	Time.	Steam-pressure.	Condensing-water. Tare.	Condensing-water. Gross weight.	Condensing-water. Net weight.	Wet steam. Gross weight.	Wet steam. Net weight.	Temperature. Cold water.	Temperature. Warm water.	Temperature. Steam.	Latent heat.				$\frac{W}{w}(t - t') - (T - t)$	Per cent of entrained water.	Quality of steam.	Degrees of superheat.

THROTTLING CALORIMETER.

Quality of steam $x = \dfrac{H - t + 0.48\,(T - T')}{l}$. Degrees superheat $S = \dfrac{x - 1}{0.48}$.

		P	p	T	t'	T'	$T-T'$	$.48(T-T')$	H	L	$H-t$		$1-x$	x	S
Number.	Time.	Steam-pressure. Main.	Steam-pressure, Calorimeter.	Observed temp. Calorimeter.	Temp. steam in main.	Normal temp. in Calorimeter.	Degree superheat.	Thermal units superheat.	Total heat, pressure p.	Latent heat for pressure P.		$H - t + .48\,(T - T')$	Per cent of entrained water.	Quality of steam.	Degrees of superheat.

Carpenter's calorimeter, in its simplest form, consists of two vessels about 8 inches long and 2 inches in diameter, supplied with a water-gauge connection in the lower portion, and having an attached scale graduated to read into hundredths of a pound of water. Each chamber constitutes a separating device; and the steam from the first passes into the second through an orifice sufficiently large to maintain the same pressure in both chambers, and from the second it is discharged by a smaller orifice into the air. The reading at the first chamber gives the total weight, W, of water in the steam, plus the loss by condensation u. In the second chamber the reading is the condensation-loss u, since it is supplied with dry steam. In use, the discharged steam is condensed and weighed, or computed, giving the value of w; the water deposited is noted from time to time in the two chambers, giving respectively W and u. These determinations are reduced to pounds per hour in each case, and the moisture is computed by the formula

$$1 - x = \frac{W - u}{W + w}, \quad \text{or} \quad x = \frac{w + u}{W + w}.$$

Experience indicates that it is practicable to determine the quality of steam accurately to within one quarter of one per cent. There is no limit so far as moisture is concerned, but the amount of superheat that can be determined depends upon the value of u, which in a number of cases has been about two per cent.

All thermometers should be calibrated for boiling-point or freezing-point, and should be tested for air-bubbles.

The methods of obtaining the sample of steam lead to more error and cause more uncertainty than all the other errors combined.

The standard calorimeter connection is an unperforated tube extending into the main pipe one-half inch from the side.

180. Sampling the Flue Gas should be carried on at intervals of 10 to 15 minutes throughout the trial. The gas should be received in an air-tight pipe or jar. The composition of the gases should be determined as far as regards carbonic acid, carbonic oxide, and oxygen. The tube should be of porcelain or glass for very hot flues, since iron tubes at such temperatures are oxidized. Supposing an analysis of the gas gives K per cent of carbonic acid, O per cent of oxygen, and N per cent of nitrogen, then the proportion of air actually used to the theoretical quantity required is 1 to x.

Where

$$x = \frac{N}{N - \frac{79}{21}O} \quad \text{or} \quad \frac{21}{21 - 79\frac{O}{N}},$$

unity of weight of this coal will then give, at a temperature of 0° and a pressure of one atmosphere,

$$\frac{1854}{10}C = \text{carbonic acid};$$

$$\frac{KO}{K} = \text{oxygen};$$

$$\frac{KN}{K} = \text{nitrogen}.$$

The quantity of moisture in the escaping gases may be calculated from the moisture in the coal, from that formed by burning the hydrogen, and from that contained in the air admitted to the furnace where the latter has been determined. Any serious break in the setting can be detected by filling the grate with smoky coal and then closing the damper.

The apparatus and methods are described in chemists' laboratory reference-books.* The reactions are the following:

CO_2 is acted on thus: $2KOH + CO_2 = K_2CO_3 + H_2O$, the salt formed being dissolved by the water present. The volume of remaining gas is measured under atmospheric pressure.

The difference between this volume in cubic centimetres and the original 100 gives the percentage of carbonic-acid gas present.

The remaining gas is treated in a similar manner with a solution consisting of ten per cent KOH and five per cent solution of potassium pyrogallate, for the determination of free oxygen. Free oxygen is absorbed readily by a strong solution of pyrogallic acid ($C_6H_6O_3$) in caustic potash (KOH); the reaction being as follows: $C_6H_3(OH)_3 + 12O = 6CO_2 + 3H_2O$ and $6CO_2 + 12KOH = 6K_2CO_3 + 6H_2O$.

Carbon dioxide and oxygen removed, the remaining gas is composed of carbon monoxide and nitrogen. The carbon monoxide may be determined by treatment with cuprous chloride dissolved in concentrated hydrochloric acid, forming the compound ($CO,Cu_2Cl_2,2H_2O$), which would afford a check on the results not obtained otherwise. The data and results on pages 623 and 624 are for a sample case of this kind.

181. **Draught-gauges** are made for the purpose of determining the head producing draught and the intensity of the draught. They are of many forms, all of which usually depend upon the measurement of the head of water which balances that head at the chimney. A very compact and accurate form of draught-gauge, used by the Author with very satisfactory results, is that of Mr. J. M. Allen (Fig. 193).

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* See also Thurston's Engine and Boiler Trials; § 47.

FLUE-GAS ANALYSIS.

Air being equal to four parts of Nitrogen plus one part of Oxygen by volume.

	Symbol.	Formula.	Determination.		
			1	2	3
Reading Draught Gauge, water inches	r	....	.4	.4	.4
Temperature Flue	T	....	300°	300°	320°
Temperature Boiler-room	t	....	76°	76°	76°
Temperature Outside Air	t'	....	42°	42°	42°
Weight of Coal burned per second	w	....	.5 lbs.	.5 lbs.	.5 lbs.
CO_2, per cent of volume	K'	....	6.9	6.8	7.6
$O + CO_2$, per cent of volume		....	16.5	16.9	16.9
O, free oxygen, per cent of volume	O'	....	9.6	10.1	9.3
CO, per cent of volume	U'	$\frac{1}{3}[100-(S+1)(K'+O')]$	1.98	2.07	1.91
Nitrogen, per cent of volume	N'	$\frac{2}{3}\left[100+\left(\frac{S}{2}-1\right)(K'+O')\right]$	77.6	77.9	77.9
Dilution coefficient	X	$\frac{N'}{N'-4O'}$	1.85	2.1	1.9
Proportional weight	M	$14N'+16O'+28U'+44K'$	1706.	1694.2	1713.1

	Symbol.	Formula.	Determination. 1	2	3
Per cent free O. by weight		$\frac{16O'}{M}$	9	9.5	8.6
Per cent total O. by weight		$\frac{16}{M}(2K' + U' + O')$	27.4	23.8	27.5
Per cent total carbon by weight		$\frac{12}{M}(U' + K')$	8.9	8.4	8.8
Weight of air per pound carbon	W	$12X$	22.5	24.8	22.9
Heat-units lost	B	$0.238(T - t)W$	1210.18	1322.1	1329.8
Efficiency, per cent	e	$\frac{14500 - B}{14500}$	91.6	90.8	80.8
Volume air per pound coal at 32° F.	Vo	$12.5 W$	283.7	310.	286.2
Velocity in feet per second	u	$\frac{wV_o(461 + T)}{493A}$	17.8	19.8	18.6
(From draught-gauge.) Head to produce draught	h'	$\frac{(461 + T)r}{94\ 7\left(.0807 + \frac{1}{V_o}\right)}$	38.6	38.3	39.
Actual height	H'		100	100	100
Throttling effect damper		$H' - H$	61.4	61.7	61.0

A and *A'* are glass tubes, mounted as shown, communicating with each other by a passage through the base, which may be closed by means of the stop-cock shown. Surrounding the glass tubes are two brass rings, *B* and *B'*. These rings are attached to blocks which slide in dovetailed grooves in the body of the instrument, and may be moved up and down by screws at *F*, *F'*. The scales are divided into fortieths of

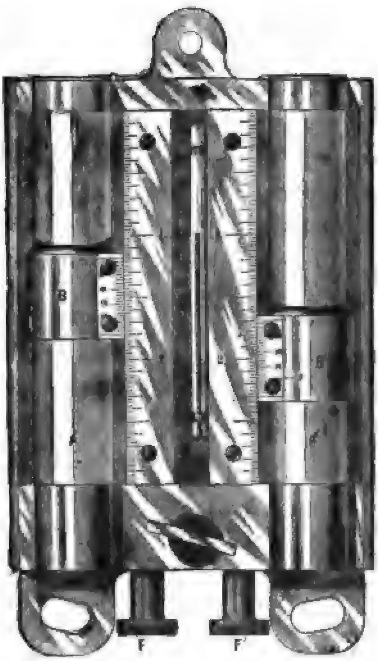

FIG. 193.—DRAUGHT-GAUGE.

an inch, and read to thousandths of an inch by the verniers *e* and *e'*, which are attached to the sliding-rings *B*, *B'*. If the two short rings are set at different heights, the difference in readings will give the difference of level. The thermometer is for the purpose of noting the temperature of the external air.

182. Standard Methods of Engine Trial have been proposed with the double purpose of securing all needed data, at least cost in time and money, and of making all results strictly

comparable. All such experimental work, whether for directly practical purposes or with a scientific object, should be made by that carefully planned and precisely stated system which should be best adapted to the ready determination of all needed data, with least liability to error, and a most convenient means of checking all figures. Both engine and boiler trials should be made in accordance with such a system as is generally recognized as well adapted to its purposes.

The results of any engine-trial, if complete and accurate, should enable the engineer to answer several questions:

(1) What is the real efficiency and the economical performance of the system tested?

(2) How does it compare with standard apparatus of a similar character? and in what is it superior or inferior? What are its excellencies and its defects?

(3) How are commercial and financial conditions affected by its operation?

These questions being solved, the proprietor will know to what extent his expenditures and his methods of operation are wise and productive; the builder will learn how successful he has been in his work, where it is defective, and what remedy is available; the engineer secures data which enable him to design intelligently later and better constructions, and it may furnish a standard for still other comparisons.

The duty-measurement should always be expressed in perfectly definite terms. If the efficiency of the system, engines and boilers included, is to be measured only, it is sufficient to ascertain the relation between the work performed and the cost of its performance as measured at the boilers; but a system of measurement which determines the heat produced in the furnace, in thermal units, or other equally definite terms, and the quantity of *useful* work which it yields, is the only satisfactory one.

The only correct and exact method of gauging the performance of any steam-engine is to determine the weight of steam, or, better, the number of thermal units, demanded by it per horse-power per hour under specified conditions. The proper

measure of the boiler-efficiency is the proportion of the heat, of combustion of the fuel, which is absorbed and stored as available energy in the steam which it produces. To rate the engine by the quantity of fuel burned in its boiler is wholly incorrect; to rate the boiler by the ratio of steam made to coal burned is hardly less indefinite. It is only by the habitual use of a known fuel of uniform composition and physical character that comparisons of value may be effected at all. Even where the steam-unit is adopted, it must be taken at a standard temperature and pressure. In all heat-engines the proper measure of heat-energy is the heat-unit. There is therefore reason in the adoption of, for an example, 1,000,000 B. T. U., a figure sometimes so taken, as a standard quantity in duty-trials of engines.*

183. Two Methods of Trial are available in testing steam-engines, both of which are found to be capable of giving exact results: (1) Measuring the energy supplied by the boiler in the form of heat transferred to the engine by the steam, and comparing the mechanical equivalent of this heat-energy with the quantity of mechanical energy obtained from the engine. (2) Determining the amount of energy rejected, as measured in the heat carried away by the exhaust, and similarly comparing this with the work done. In the first case, the quotient of the useful energy gained by the total energy expended is a measure of the efficiency of the system; in the second case, the same measure is obtained by dividing the work done by the sum of that quantity and the rejected energy. Of these two systems of trial, the first is that customarily employed by engineers for many years past; the second is that comparatively recently introduced by Messrs. Farey and Donkin.

The first system being adopted, the quantity of heat-energy expended is measured by determining the weight of steam produced and its physical condition, and the quantity of heat brought to the boiler by the feed-water. The total heat com-

* This corresponds to 100 lbs. coal evaporating 11.25 lbs. water per lb., from and at 212° F.

municated to the steam, less the heat received with the feed, is the net expenditure. It is usual to take a standard temperature, as 0° F., 32° F., or 0° C., as that to which all temperature measurements are referred. In such case, assuming the standard point on the scale to be 0°, the total heat supplied by the boiler is ascertained by weighing the feed-water for a specified time, and thus determining the weight of steam, wet or dry, passing to the engine; next ascertaining what proportion of the fluid is still liquid, or what is the amount of superheating; computing the heat stored in the fluid; then, finally, deducting the heat stored in the feed-water, both measured from 0°, thus obtaining the net quantity which comes from the fuel.

The second system being employed, the quantity of rejected heat is determined by measuring that received in the condenser and wasted in other ways. The total rejected heat consists of the following parts: (1) Heat carried away by air and vapor from the hot-well and by the water of condensation, measured from 0° or the standard point on the thermometer. (2) Heat received and carried away by the condensing-water, the measurement being made between the limits of reception and rejection of that water. (3) Heat wasted by conduction and radiation from the exterior of the heated parts of the machine.

In illustration of such distribution of energy we find the following, as deduced by Prof. Ewing* from data supplied by Mr. Main: †

Data.

Steam-pressure, absolute, lbs. per sq. in	76
Time occupied by trial, hours	6
I. H. P	127.4
Feed-water, lbs. per revolution	1.394
Air-pump discharge, lbs. per revolution	51.1
Water drained from jackets, lbs. per revolution	0.186
Per cent priming	4
Temperatures: feed, injection, and discharge	59°, 50°, 73°.4

* Encyclopedia Britannica, 9th ed.; art. Steam-engine.

† Minutes Proc. Inst. C. E., vol. LXX.

Results.

Quality of steam	0.96
Quantity of steam supplied per revolution, lbs	1.028
" " injection-water " " "	49.9
Latent heat of steam, B. T. U	898
Heat in water of boiler, "	278°
" " " " feed, "	27
" " " " injection, B. T. U	18
" " " " discharge, "	41.4
Heat from boiler to engine, per revolution	1377
" " " " jackets, " "	212
" " " total B. T. U. per revolution	1589
" returned to boiler, " " "	38
" *net* supply " " "	1551
" converted into work " " "	227
" total rejected " " "	1324

The loss by conduction and radiation, externally, was about 6 per cent. The actual efficiency of the engine was

$$E = \frac{227}{1551} = 0.146,$$

or not quite 15 per cent, while the thermodynamic efficiency was 0.335, more than twice as great.

In duty-trials of pumping-engines, the best system yet proposed is probably that which bases the efficiency determination upon the measured amount of work done by the system on a consumption of 1,000,000 B. T. U. The heat consumed should be taken to be all supplied by the fuel, or all received by the engine, including that wasted by all its accessories.

The useful work should be, wherever practicable, measured by the product of weight of water pumped, as ascertained by the use of a weir, into the head against which it is pumped, as measured by a pressure-gauge, or otherwise, at the pump-delivery. Losses by leakage, lost action, etc., are thus detected. Internal friction thus properly tells against the engine; external friction—in mains, etc.—is as properly ignored.

184. The Farey and Donkin System of trial of engines is one in which the quantity of heat supplied by the boiler and

received by the engine is not directly determined, but is ascertained by observation of the quantities of heat rejected by the engine and carried away in the condensing water. This method only applies to condensing-engines and to those which can be temporarily converted into condensing-engines for the purposes of the test. A boiler-trial is always a troublesome and disagreeable operation, and usually involves considerable expense both in preparation and in its conduct. Where it is only the engine that is to be tried and judged, the avoidance of a boiler-trial is a decided advantage. The ability to test an engine by itself is very often an important desideratum, and especially as permitting more frequent determinations of the condition of the machine and a more complete knowledge of its action at all times.

It has been seen that the heat supplied to any engine is disposed of in three directions: by conduction and radiation to surrounding objects, by conversion into mechanical work, and by rejection in the exhaust-steam and the water accompanying it. Of these quantities the first is comparatively small, and is often entirely ignored as unimportant; the second ranges in good engines between, perhaps, 10 and 15 per cent, rarely exceeding the latter figure; while the last item includes, as a rule, above 85 per cent, and generally 90 per cent, of the total quantity sent over from the boilers. In the condensing-engine all this heat may be found and measured up in the water passing out at the delivery-pipe from the hot-well. It is obvious that the sum of this heat-equivalent of the indicated power of the engine, plus the heat so rejected, and the small quantity added to represent losses by radiation and conduction, will be the measure of the heat-supply from the boiler. To determine this total, therefore, we have but to measure the indicated power of the engine and the heat discharged from the condenser. The first of these processes is already understood, To secure the second measurement, it is only necessary to measure the flow of the heated water by a weir and notch, at the same time measuring its temperature by accurate thermometers. A high value for the quantity of heat discharged.

per horse-power and per hour, indicates an inefficient engine; a low value is the proof of good economy.

According to Clark, Mr. John Farey, as early as 1849, gauged the efficiency of the steam-engine by measuring the quantity of its rejected heat.* Hirn made the system practically applicable, and initiated more general use, beginning about 1855.† Messrs. Farey and Donkin introduced the now familiar system, and published their own earliest results in 1868.‡ This method will be fully illustrated later.

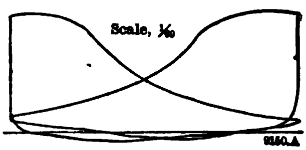

FIG. 194.—INDICATOR-DIAGRAM.

Professor Unwin has incorporated such a method in a standard system of trial of the Worthington engine with the "equalizers" attached. The following are the reported results of this very complete work: §

The Engines.—The engines are compound, pumping a large volume of water on a low lift. The high-pressure pistons are 27 in. in diameter, and the low-pressure pistons 54 in. The stroke is variable, the maximum from cylinder-head to cylinder-head being 44 in. During the trials the stroke remained very constant, and about 43 inches. The engines work rams 40 in. in diameter, and the same stroke as the steam-pistons. Compensating cylinders absorb work during the first half of the stroke and give it back during the second half. There are two to each engine, 11 in. in diameter, loaded by

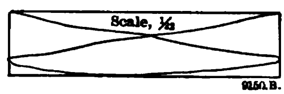

FIG. 195.—DIAGRAMS.

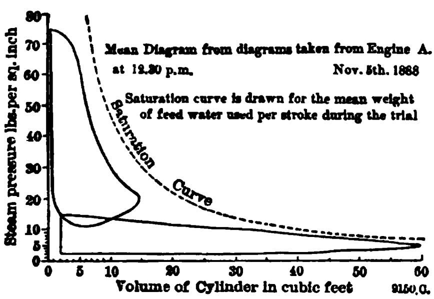

FIG. 196.—MEAN DIAGRAMS.

* Steam-engine, vol. I., p. 585.

† Bulletin de la Soc. de Mulhouse, 1857; p. 5.

‡ Engineering, Feb. 15, 1884, p. 154; Ibidem, July 17, 1868, May 21, 1869.

§ London Engineering, December 7, 1888. For full report see also Thurston's Engine and Boiler Trials; § 103, p. 424.

air-pressure to about 120 pounds per square inch. The pumps lift water from a well communicating with the river and deliver it through two 3-ft. mains to the reservoirs, nine miles distant. The head during the trials, measured by the difference of pressure in the suction and discharge pipes, was from 50 ft. to 65 ft. The head was measured by mercury-columns fixed in the engine-house, communicating with the suction and delivery mains in accordance with the provision of the contract.

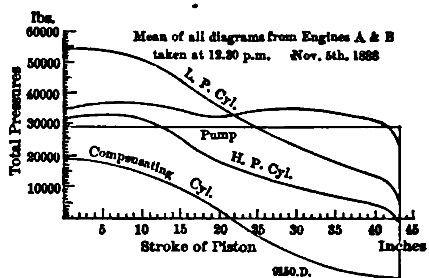

FIG. 197.—PRESSURE-CURVES.

The suction-gauge communicated with the suction-pipe just

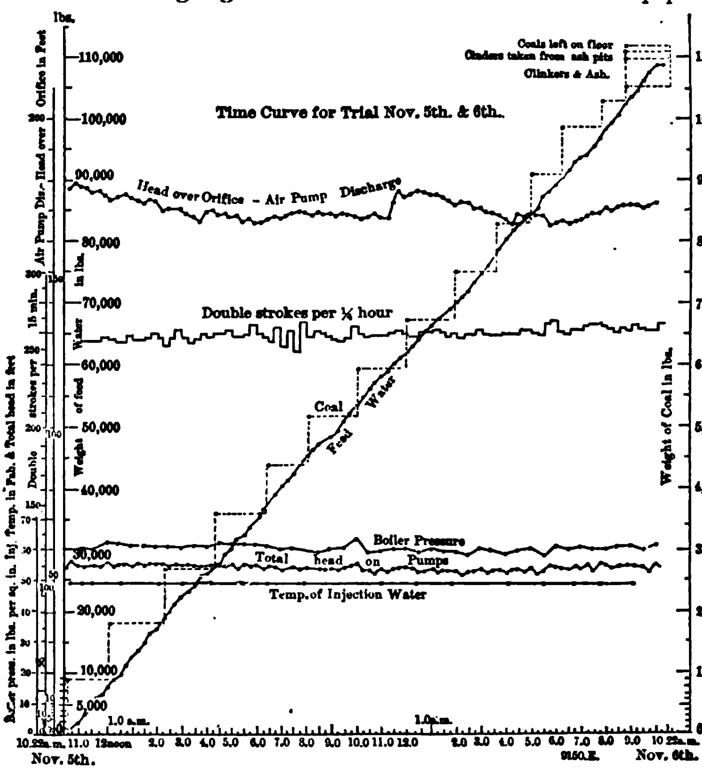

FIG. 198.—CHART OF TRIAL.

below the floor, and the pressure-gauge was set to give pressures reckoned from the floor-level. The sum of the mercury-gauge readings is taken as the effective lift. The pressure-gauge communicates with the delivery-main at a point beyond the stop-back valve. Consequently the resistance of that valve is reckoned as part of the engine-friction, and is not credited to the useful work done by the pumps.

The engine-cylinders are jacketed, and the steam is also taken through a jacketed reservoir between the cylinders. The jacket-water was weighed. The condensers are injection-condensers with horizontal air-pumps.

The Boilers.—The boilers were single-flued Cornish boilers, 28 ft. in length and 6 ft. in diameter, with a single flue 3 ft. 6 in. in diameter for the greater part of the length.

During the trials the grate-area was 60 square feet.

Measurement of the Feed.—The feed was supplied from the delivery-main at a nearly constant temperature of 51 deg., the ordinary feed-arrangements which supply the boilers with hot water from the jackets and hot-well being disconnected. The feed was delivered into a small gauge-tank with overflow-pipe provided with a float and counter. The capacity of this gauge-tank was determined three times by weighing the water; and the closely accordant measurements gave a mean value of 394 pounds for the capacity.

The feed-tank delivered by a stop-valve into another tank, from which a small Worthington, feed-pump delivered the water into boilers.

The Worthington pump took its steam from the boilers in use, and exhausted into the tank, from which it pumped. The whole of the steam used was therefore recondensed and returned to the boilers.

Of the heat supplied by the boilers to work the feed-pump, nearly all was returned to the boilers. A small portion, viz., that due to the useful work of pumping and that lost by radiation from the tank, was no doubt lost. So far a small error telling against the main engines is introduced.

The water-level at the commencement of each trial in the

boiler gauge-glasses was carefully observed, and the water-level was brought to exactly the same marks at the end of the trials. The time at which each tankful was supplied to the boilers was noted, and also the feed-water temperature. Pyrometer observations were made in the flues. Anemometer observations of the air supplied to each boiler were taken every half-hour during the twenty-four hours' trial, the anemometer having been previously tested.

Measurement of the Air-pump Discharge.—The air-pump discharge was led into a wooden tank with stilling-screens. From this it was discharged through a sharp-edged circular orifice freely into the air. The diameter of the orifice was carefully tested after the trials, and the coefficient of discharge from similar orifices is known to be 0.599. The temperature and head over the orifice was noted every 5 minutes in the first trial and every 7½ minutes in the second. The temperatures relied on in this report were taken by a fixed zero thermometer, with open scales, recently verified at Kew.

Measurement of Length of Stroke.—As the stroke is variable, an arrangement of indicating-fingers was attached to each engine, and the length of stroke on each engine was noted every quarter of an hour.

Indicated Power.—The indicated power was taken by four indicators, chosen because they give fairly large diagrams. These indicators were tested under steam, against a steel-tube pressure-gauge recently made, and specially tested. The indicator-pipes were large and were clothed. Diagrams were taken every half-hour from all the cylinders.

185. **The Scheme of the Trial** should be carefully prepared in advance, and should be so planned as to secure the needed data with certainty and accuracy. The first consideration is the purpose of the proposed trial, and the first work done the arrangement of a general plan that shall enable the observers to collect with accuracy and certainty all the needed data, and to record them conveniently and in most available form. The next matter to be studied is the reduction of all general and special operations of the trial to a complete and

efficient system, in which every part shall be made so far as possible contributory to the efficiency and fruitfulness of every other part; in which each observer shall be so stationed and so instructed that he may secure the data assigned him for collection with least difficulty, risk, and uncertainty, and shall have his own work checked, and shall aid in checking the work of others, as completely as possible. No essential data should remain unchecked, and every subsequent calculation based upon them should also be made by at least two computers independently.

The plan of the work being settled upon, each detail should be studied by itself, and every provision that experience and foresight can suggest should be taken to insure perfection of the scheme. A preliminary and informal trial will then be likely to reveal any serious defect, which being corrected, the final and official trial may be fully expected to give thoroughly reliable results.

Instructions for Tests of Boilers for the U. S. Navy are as follow: *

GENERAL.

All dimensions and weights that may be obtained from drawings or by information will be verified, as far as possible, by actual measurement or weight. Detailed drawings, whenever they can be obtained, with the verified dimensions noted on them, will accompany the report of the Board.

The necessary instruments for the tests will be furnished by and returned to the Bureau of Steam-engineering.

The report, accompanied by a separate letter of transmittal, will be written in copying-ink, and submitted to the Bureau of Steam-engineering, the press copy to accompany the original.

EVAPORATIVE TESTS.

Before beginning the test, the boilers will be inspected to see that they are perfectly clean internally, and without leaks.

* Report of Bureau Steam-engineering, 1890.

If small leaks unavoidably occur, the loss in water and heat will be accurately accounted for in the computations. All water supplied to the boiler will be carefully measured by means of two tanks, suitably connected and arranged, whose capacities have previously been accurately obtained, either by measurement or weight, or both.

The temperature of the water in these tanks will be taken at the top, middle, and bottom, as the water is pumped out. The time it takes to empty each tank will also be noted. When cold feed-water must be used, it should enter the tanks at the top and opposite to the suction of the feed-pump.

The fuel consumed during the test will be carefully weighed, the amount supplied being carefully noted at the receiving and checked at the delivery end. The weight of the dry refuse from it, in ashes, dust, and clinker, if any, will also be accurately ascertained and noted. When, in starting the second fire, wood and other substances are used, their value as fuel will be allowed for in the total quantity of combustible comsumed at the rate of $\frac{4}{10}$ pound of coal per pound of wood. Then, from this quantity and the percentage of the dry refuse from fuel, the total quantity of fuel consumed will be calculated.

A statement of the kind and quality of the fuel, when and where obtained, and chemical analysis, if one can be or has been made, will be embodied in the report.

The data called for by form No. 80–1 will be accurately taken every thirty minutes. The steam-pressure will be kept as steady as possible at 160 pounds by gauge, and regulated by means of the stop-valve.

After steam has been raised to the working pressure, the fires will be hauled and the grates and ash-pits quickly cleaned. Fires will then immediately be started with weighed fuel, the height of the water in the boilers carefully marked, the first full tank connected with the boiler through the feed-pump, the time noted, and the test considered as having commenced. During the test, the operation of the boiler and all other apparatus in use will be carefully and minutely observed and noted.

The evaporative test will continue for twenty-four consecu-

tive hours. Should anything occur to interfere with the continuance of the test for this length of time, another test under these instructions will be commenced as soon as practicable.

During this test the forced-draught will be maintained at a pressure equal to 2 inches of water at the mouth of the ash-pit.

At the end of the test the conditions of fire and water level will be, as nearly as possible, the same as at the beginning.

CALORIMETRIC TESTS.

During the trial frequent calorimetric tests of the steam will be made. The calorimeter should have at least 40 gallons capacity, and be free from leaks. The platform-scales should be carefully tested, and should be accurate to within $\frac{1}{100}$ of a pound. Steam will be taken by a well-covered 1-inch pipe from the centre of the main steam-pipe, as near the boiler as possible, and led to the calorimeter. The pipe at the calorimeter end will be fitted with the short-section of steam-hose provided.

A high-grade thermometer will be fitted in the steam-pipe as near as possible to the 1-inch pipe. Before turning the steam into the calorimeter, the pipe must be blown through sufficiently long to clear it of all condensed steam and to heat it thoroughly.

The water admitted to the calorimeter should be of the lowest temperature obtainable, and the steam blown into it long enough to raise its temperature to about 11 degrees. The water should be continually agitated until the end, to insure a uniform temperature.

The openings in the top of the calorimeter will be as small as possible, in order that evaporation may be reduced to a minimum.

A very small drain-cock will be fitted to the bottom of the calorimeter.

The data of the calorimetric tests will be accurately obtained and noted on the blank form No. 80–3, and the computations from them carefully made.

SPECIAL TESTS.

The effect on the water in the boiler and on the fires and gases of combustion, under various pressures of forced draught (up to the highest safe limit), and the effect of suddenly closing the boiler stop-valve and opening it again after ten minutes. will be made the subject of special experiment after the evaporative test has been completed.

The report will include a general description of the hull, if the boilers are in a vessel, with such dimensions and facts as may be thought necessary or useful; a detailed description of the engines, if the boilers are in a vessel, with such weights, dimensions, and data of space occupied as can be obtained; details of propellers, if any; detailed description of the boilers and all special appliances with which these may be fitted, with all weights and dimensions, cubic space occupied by boilers, and space necessary around boilers to facilitate repairs, weight when empty, and with water to steaming level, the facility for repairs and renewal of parts, and liability of derangement; and the class of skilled labor required for the proper care of boiler when in use and for the necessary repairs.

The subject-matter of the report will be arranged in the following order:

1. Preliminary statement of board in relation to carrying out instructions and arrangements made for the test.

2. Description and data of hull.

3. Description and data of engines.

4. Description and data of propellers.

5. Description and data of boilers and special appliances.

6. Fuel used.

7. Description of the manner of making the evaporative tests.

8. Data of these tests (form No. 80–1).

9. Description of the manner of making the calorimetric tests.

10. Data and computations of above (forms No. 80–2, 3, and 4).

12. Conclusions. Deductions from the results of the tests, and observations on the points referred to in the general and special instructions to the board, and such other observations or suggestions as the board may think necessary or useful.

Appendix A. Copy of orders convening board.

Appendix B. Copy of bureau's detailed instructions to the board.

Appendix C. Drawings, in the order of the subjects as previously given. Such other information as may be appended will follow Appendix C.

186. Competitive Trials of Engines are sometimes conducted by the engineer, either to determine which of two or more competing forms of engine is to be accepted by the purchaser, or by his client, or, as at exhibitions of various kinds, simply to ascertain the power and efficiency of two or more engines, with a view to deciding their relative merits as types of engine, or as representing the best practice of their builders. It is largely through this kind of competition that the best-known systems of standard engine-trial have been developed.

187. A Standard Engine-trial, conducted in accordance with the recommendations of a committee of the American Society of Mechanical Engineers, is carried out under the following instructions as nearly as practicable, this plan being especially that proposed for pumping-engines: *

(1) TEST OF FEED-WATER TEMPERATURES.

The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of three hours, or such a time as is necessary to find the temperature of the feed-water (or the several temperatures, if there is more than one supply) for use in the calculation of the duty. During this test observations of the temperature are made every fifteen minutes. Frequent observations are also made of the speed, length of stroke, indication of water-pressure gauges, and other

* Trans. Am. Soc. M. E., vol. XI, No. CCCLXXX; 1890.

instruments, so as to have a record of the general conditions under which this test is made.

Directions for obtaining Feed-water Temperatures.—When the feed-water is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed-pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperature of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejected heat of the engine, such as that contained in the exhaust-steam either of the main cylinders or of the auxiliary pumps), it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feed-water supply, one the main supply from the hot-well, or from some other source, and the other an auxiliary supply derived from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed-pipe near the boiler, is to be taken at convenient points in the independent pipe.

When a separator is used in the main steam-pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot-well, or to the jacket-tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket-tank, this tank being subjected to boiler-pressure, the

temperature of the separator-water and jacket-water are each to be taken before their entrance to the tank.

Should there be any other independent supply of water, the temperature of that is also to be taken on this preliminary test.

Directions for Measurement of Feed-water.—As soon as the feed-water temperatures have been obtained the engine is stopped, and the necessary apparatus arranged for determining the weight of the feed-water consumed, or of the various supplies of feed-water if there is more than one.

In order that the main supply of feed-water may be measured, it will generally be found desirable to draw it from the cold-water service-main. The best form of apparatus for weighing the water consists of two tanks, one of which rests upon a platform-scale supported by staging, while the other is placed underneath. The water is drawn from the service-main into the upper tank, where it is weighed, and it is then emptied into the lower tank. The lower tank serves as a reservoir, and to this the suction-pipe of the feeding apparatus is connected.

The jacket-water may be measured by using a pair of small barrels, one being filled while the other is being weighed and emptied. This water, after being measured, may be thrown away, the loss being made up by the main feed-pump. To prevent evaporation from the water, and consequent loss on account of its highly heated condition, each barrel should be partially filled with cold water previous to using it for collecting the jacket-water, and the weight of this water treated as tare.

When the jacket-water drains back by gravity to the boiler, waste of live steam during the weighing should be prevented by providing a small vertical chamber, and conducting the water into this receptacle before its escape. A glass water-gauge is attached, so as to show the height of water inside the chamber, and this serves as a guide in regulating the discharge-valve.

When the jacket-water is returned to the boiler by means of a pump, the discharge-valve should be throttled during the test, so that the pump may work against its usual pressure,

that is, the boiler-pressure as nearly as may be, a gauge being attached to the discharge-pipe for this purpose.

When a separator is used and the entrained water discharges either to waste, to the hot-well, or to the jacket-tank, the weight of this water is to be determined, the water being drawn into barrels in the manner pointed out for measuring the jacket-water. Except in the case where the separator discharges into the jacket-tank, the entrained water thus found is treated, in the calculations, in the same manner as moisture shown by the calorimeter-test. When it discharges into the jacket-tank, its weight is simply subtracted from the total weight of water fed, and allowance made for heat of this water lost by radiation between separator and tank.

When the jackets are drained by a trap, and the condensed water goes either to waste or to the hot-well, the determination of the quantity used is not necessary to the main object of the duty trial, because the main feed-pump in such cases supplies all the feed-water. For the sake of having complete data, however, it is desirable that this water be measured, whatever the use to which it is applied.

Should live steam be used for reheating the steam in the intermediate receiver, it is desirable to separate this from the jacket-steam, if it drain into the same tank, and measure it independently. This, likewise, is not essential to the main object of the duty trial, though useful for purposes of information.

The remarks as to the manner of preventing losses of live steam and of evaporation, in the measurement of jacket-water, apply to the measurement of any other hot water under pressure, which may be used for feed-water.

Should there be any other independent supply of water to the boiler, besides those named, its quantity is to be determined independently, apparatus for all these measurements being set up during the interval between the preliminary run and the main trial, when the plant is idle.

(2) THE MAIN DUTY-TRIAL.

The duty-trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler as well as that of the engine. The test of the two will go on simultaneously after both are started, but the boiler-test will begin a short time in advance of the commencement of the engine-test, and continue a short time after the engine-test is finished. The mode of procedure is as follows:

The plant having been worked for a suitable time under normal conditions, the fire is burned down to a low point and the engine brought to rest. The fire remaining on the grate is then quickly hauled, the furnace cleaned, and the refuse withdrawn from the ash-pit. The boiler-test is now started, and this test is made in accordance with the rules for a standard method recommended by the Committee on Boiler Tests of the American Society of Mechanical Engineers. This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler having previously been heated to its normal working degree; operating the boiler in accordance with the conditions determined upon; weighing coal, ashes, and feed-water; observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam; and at the close of the test hauling the fire, and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam-pressure being the same as at the time of lighting the fire at the beginning of the test.

Previous to the close of the test it is desirable that the fire should be burned down to a low point, so that the unburned coal withdrawn may be in a nearly consumed state. The temperature of the feed-water is observed at the point where the water leaves the engine heater, if this be used, or at the point where it enters the flue-heater, if that apparatus be employed. Where an injector is used for supplying the water, a deduction

is to be made in either case for the increased temperature of the water derived from the steam which it consumes.

As soon after the beginning of the boiler-test as practicable the engine is started and preparations are made for the beginning of the engine-test. The formal commencement of this test is delayed till the plant is again in normal working condition, which should not be over one hour after the time of lighting the fire. When the time for commencement arrives the feed-water is momentarily shut off, and the water in the lower tank is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass of the boiler, the indication of the counter on the engine, and the time of day; after which the supply of feed-water is renewed, and the regular observations of the test, including the measurement of the auxiliary supplies of feed-water, are commenced. The engine-test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter are observed, together with the number of tanks of water thus far supplied, and the engine-test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling, and completing the boiler-test. It is then stopped, and the final observations relating to the boiler-test are taken.

The observations to be made and data obtained for the purposes of the engine-test, or duty-trial proper, embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner pointed out. They also embrace the number of hours' duration, and number of single strokes of the pump during the test; and, in direct-acting engines, the length of the stroke, together with the indications of the gauges attached to the

force and suction mains, and indicator-diagrams from the steam-cylinders. It is desirable that pump-diagrams also be obtained.

Observations of the length of stroke, in the case of direct-acting engines, should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments—such as steam-gauge, vacuum-gauge, thermometer in pump-well, thermometer in feed-pipe; thermometer showing temperature of engine-room, boiler-room, and outside air; thermometer in flue, thermometer in steam-pipe, if the boiler has steam-heating surface, barometer, and other instruments which may be used—every half-hour. Indicator-diagrams should be taken every half-hour.

When the duty-trial embraces simply a test of the engine, apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled, and the special observations relating to the performance of the boiler will not be taken.

Directions regarding Arrangement and Use of Instruments, and other Provisions for the Test.—The gauge attached to the force-main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome, and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force-main. By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the centre of the gauge, and correction for this element of variation is avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two

ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact scale-marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used, which acts directly on the moisture tested. If either the superheating calorimeter* or the wire-drawing† instrument be employed, the steam which it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface-condenser of some simple construction, and measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations the specific heat of the material of the barrel or tank should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam-pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator-pipes and connections used for the water-cylin-

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* Vol. VII, p. 178; 1886.

† Vol. XI, 1890, Transactions A. S. M. E. (Paper on "A Universal Calorimeter," May, 1890).

ders should be of ample size, and, so far as possible, free from bends. Three-quarter-inch pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator-springs which are correct under steam heat are erroneous when used for cold water. When such springs are used, the actual scale should be determined, if calculations are made of the indicated work done in the water-cylinders. The scale of steam-springs should be determined by a comparison, under steam-pressure, with an accurate steam-gauge at the time of the trial, and that of water-springs by cold dead-weight test.

The accuracy of all the gauges should be carefully verified by comparison with a reliable mercury-column. Similar verification should be made of the thermometers, and if no standard is at hand, they should be tested in boiling water and melting ice.

To avoid errors in conducting the test, due to leakage of stop-valves either on the steam-pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

(3) LEAKAGE-TEST OF PUMP.

As soon as practicable after the completion of the main trial (or at some time immediately preceding the trial) the engine is brought to rest, and the rate determined at which leakage takes place through the plunger and valves of the pump, when these are subjected to the full pressure of the force-main.

The leakage of the plunger is most satisfactorily determined by making the test with the cylinder-head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and

the water from the force-main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

Should the escape of the water into the engine-room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow-pipe in the wooden head. The apparatus may be constructed, if desired, in a somewhat rude manner, and yet be sufficiently accurate for practical requirements. The test should be made, if possible, with the plunger in various positions.

In the case of a pump so planned that it is difficult to remove the cylinder-head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction-valves, the head being allowed to remain in place.

It is here assumed that there is a practical absence of valve-leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger-test. Leakage of the discharge-valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction-valves will be shown by the disappearance of water which covers them.

If valve-leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction-valves, where there is no gate in the suction-pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump-cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge-valves of the pump.

The exact methods to be followed in any particular case, in determining leakage, must be left to the judgment and ingenuity of the person conducting the test.

(4) TABLE OF DATA AND RESULTS.

In order that uniformity may be secured, it is suggested that the data and results, worked out in accordance with the standard method, be tabulated in the manner indicated in the following scheme:

DUTY-TRIAL OF ENGINE.

Dimensions.

1. Number of steam-cylinders................................
2. Diameter of steam-cylinders................................ ins.
3. Diameter of piston-rods of steam-cylinders................ ins.
4. Nominal stroke of steam-pistons.......................... ft.
5. Number of water-plungers................................
6. Diameter of plungers.................................... ins.
7. Diameter of piston-rods of water-cylinders......... .... ins.
8. Nominal stroke of plungers.............................. ft.
9. Net area of plungers.................................... sq. ins.
10. Net area of steam-pistons.............................. sq. ins.
11. Average length of stroke of steam-pistons during trial....... ft.
12. Average length of stroke of plungers during trial............ ft.

(Give also complete description of plant.)

Temperatures.

13. Temperature of water in pump-well........................ degs.
14. Temperature of water supplied to boiler by main feed-pump. degs.
15. Temperature of water supplied to boiler from various other sources................................ .... degs.

Feed-water.

16. Weight of water supplied to boiler by main feed-pump....... lbs.
17. Weight of water supplied to boiler from various other sources. lbs.
18. Total weight of feed-water supplied from all sources........ lbs.

Pressures.

19. Boiler-pressure indicated by gauge....... .... lbs.
20. Pressure indicated by gauge on force-main.................. lbs.
21. Vacuum indicated by gauge on suction-main................ ins.
22. Pressure corresponding to vacuum given in preceding line..... lbs.
23. Vertical distance between the centres of the two gauges...... ins.
24. Pressure equivalent to distance between the two gauges...... lbs.

Miscellaneous Data.

25. Duration of trial.. hrs.
26. Total number of single strokes during trial................
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating........................ % or deg.
28. Total leakage of pump during trial, determined from results of leakage-test.... .... lbs.
29. Mean effective pressure, measured from diagrams taken from steam-cylinders .... M.E.P.

Principal Results.

30. Duty .... ft.-lbs.
31. Percentage of leakage.................................... %
32. Capacity.. gals.
33. Percentage of total frictions............................ %

*Additional Results.**

34. Number of double strokes of steam-piston per minute........
35. Indicated horse-power developed by the various steam-cylinders .... I. H. P.
36. Feed-water consumed by the plant per hour...... .... lbs.
37. Feed-water consumed by the plant per indicated horse-power per hour, corrected for moisture in steam............... lbs.
38. Number of heat-units consumed per indicated horse-power per hour .... B. T. U.
39. Number of heat-units consumed per indicated horse-power per minute...................... .... B. T. U.
40. Steam accounted for by indicator at cut-off and release in the various steam-cylinders............................. lbs.
41. Proportion which steam accounted for by indicator bears to the feed-water consumption.................... .....

Sample Diagrams taken from Steam-cylinders.

[Also, if possible, full measurements of the diagrams, embracing pressures at the initial point, cut-off, release, and compression ; also back-pressure, and the proportions of the stroke completed at the various points noted.]

42. Number of double strokes of pump per minute..............
43. Mean effective pressure, measured from pump-diagrams...... M. E.P.
44. Indicated horse-power exerted in pump-cylinders... .... I. H. P.

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* These are not necessary to the main object, but it is desirable to give them.

Sample Diagrams taken from Pump-cylinders.

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DATA AND RESULTS OF BOILER-TEST.

[IN ACCORDANCE WITH THE SCHEME RECOMMENDED BY THE BOILER-TEST COMMITTEE OF THE SOCIETY.]

1. Date of trial.... ....
2. Duration of trial.. hrs.

Dimensions and Proportions.

3. Grate-surface wide long Area........... sq. ft.
4. Water-heating surface.................................... sq. ft.
5. Superheating-surface........... .... sq. ft.
6. Ratio of water-heating surface to grate-surface..............
   (Give also complete description of boilers.)

Average Pressures.

7. Steam-pressure in boiler by gauge......................... lbs.
8. Atmospheric pressure by barometer............... .... lbs.
9. Force of draught in inches of water....................... ins.

Average Temperatures.

10. Of steam.. degs.
11. Of escaping gases...................................... degs.
12. Of feed-water.. ....

Fuel.

13. Total amount of coal consumed* .... lbs.
14. Moisture in coal .. .. %
15. Dry coal consumed................ .... lbs.
16. Total refuse (dry).. lbs.
17. Total combustible (dry weight of coal, item 15, less refuse, item 16)................................ .... lbs.
18. Dry coal consumed per hour.............................. lbs.

Results of Calorimetric Test.

19. Quality of steam, dry steam being taken as unity............
20. Percentage of moisture in steam.......................... %
21. Number of degrees superheated... .... degs.

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* Including equivalent of wood used in lighting fire. One pound of wood equals 0.4 of a pound of coal, not including unburned coal withdrawn from fire at end of test.

Water.

22. Total weight of water pumped into boiler and apparently evaporated * .... lbs.
23. Water actually evaporated corrected for quality of steam..... lbs.
24. Equivalent water evaporated into dry steam from and at 212° F.† .... lbs.
25. Equivalent total heat derived from fuel, in British thermal units .... B.T.U.
26. Equivalent water evaporated into dry steam from and at 212° F. per hour.......... lbs.

Economic Evaporation.

27. Water actually evaporated per pound of dry coal from actual pressure and temperature.......... lbs.
28. Equivalent water evaporated per pound of dry coal from and at 212° F.......... lbs.
29. Equivalent water evaporated per pound of combustible from and at 213° F.......... lbs.
30. Number of pounds of coal required to supply one million British thermal units.......... lbs.

Rate of Combustion.

31. Dry coal actually burned per square foot of grate-surface per hour.......... lbs.

Rate of Evaporation.

32. Water evaporated from and at 212° F. per square foot of heating-surface per hour.......... lbs.

To determine the percentage of surface moisture in the coal a sample of the coal should be dried for a period of twenty-four hours, being subjected to a temperature of not more than 212°. The quantity of unconsumed coal contained in the refuse withdrawn from the furnace and ash-pit at the end of the test may be found by sifting either the whole of the refuse, or

* Corrected for inequality of water-level and of steam-pressure at beginning and end of test.

† Factor of evaporation $= \frac{H - h}{965.7}$, H and h being, respectively, the total heat-units in steam of the average observed pressure, and in water of the average observed temperature of feed.

a sample of the same, in a screen having ⅜-inch meshes. This, deducted from the weight of dry coal fired, gives the weight of dry coal consumed, for line 15.

188. Results of Actual Trial, as illustrated by the committee, would be computed by the use of the following formulæ:

$$1.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat-units consumed}} \times 1{,}000{,}000$$

$$= \frac{A(P \pm p + s) \times L \times N}{H} \times 1{,}000{,}000 \text{ (foot-pounds).}$$

$$2.\ \text{Percentage of leakage} = \frac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent).}$$

3. Capacity = number of gallons of water discharged in 24 hours

$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144}$$

$$= \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total friction

$$= \left(\frac{I.H.P. - \dfrac{A(P \pm p + s) \times L \times N}{D \times 60 \times 33{,}000}}{I.H.P.} \right) \times 100$$

$$= \left[1 - \frac{A(P \pm p \times s) \times L \times N}{A_{,} \times M.E.P. \times L_{,} \times N_{,}} \right] \times 100 \text{ (per cent);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam-piston, this last formula becomes—

$$\text{Percentage of total frictions} = \left[1 - \frac{A(P \pm p + s)}{A_{,} \times M.E.P.} \right] \times 100 \text{ (p. c.).}$$

In these formulæ the letters refer to the following quantities:

A = Area, in square inches, of pump-plunger or piston, corrected for area of piston-rod. (When one rod is used at one end only, the correction is one half the area of the rod. If there is more than one rod, the correction is multiplied accordingly.)

P = Pressure, in pounds per square inch, indicated by the gauge on the force-main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum-gauge on suction-main (or pressure-gauge, if the suction-pipe is under a head). The indication of the vacuum-gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centres of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump-well, and dividing the product by 144; or by multiplying the distance in feet by the weights of one cubic foot of water at the various temperatures.

L = Average length of stroke of pump-plunger, in feet.

N = Total number of single strokes of pump-plunger made during the trial.

A = Area of steam-cylinder, in square inches, corrected for area of piston-rod. The quantity $A_{,} \times M.E.P.$, in an engine having more than one cylinder, is the sum of the various quantities relating to the respective cylinders.

$L_{,}$ = Average length of stroke of steam-piston, in feet.

$N_{,}$ = Total number of single strokes of steam-piston during trial.

$M.E.P.$ = Average mean effective pressure, in pounds per

square inch, measured from the indicator-diagrams taken from the steam-cylinder.

I.H.P. = Indicated horse-power developed by the steam-cylinder.

C = Total number of cubic feet of water which leaked by the pump-plunger during the trial, estimated from the results of the leakage-test.

D = Duration of trial, in hours.

H = Total number of heat-units [B. T. U.] consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler-pressure reckoned from temperature of main feed-water + weight of water supplied by jacket-pump × total heat of steam of boiler-pressure reckoned from temperature of jacket-water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture, the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (i.e., the excess of the temperature of the steam above the normal temperature of saturated steam) by 0.48. No allowance is made for heat added to the feed-water, which is derived from any source, except the engine or some accessory of the engine. Heat added to the water by the use of a flue-heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a steam-reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

The following example is one of those given by the com-

mittee to illustrate the method of computation. The figures are not obtained from tests actually made, but they correspond in round numbers with those which were so obtained:

EXAMPLE.—*Compound Fly-wheel Engine.*—High-pressure cylinder jacketed with live steam from the boiler. Low-pressure cylinder jacketed with steam from the intermediate receiver, the condensed water from which is returned to the boiler by means of a pump operated by the engine. Main steam-pipe fitted with a separator. The intermediate receiver provided with a reheater supplied with boiler-steam. Water drained from high-pressure jacket, separator, and reheater collected in a closed tank under boiler-pressure, and from this point fed to the boiler direct by an independent steam-pump. Jet-condenser used operated by an independent air-pump. Main supply of feed-water drawn from hot-well and fed to the boiler by donkey steam-pump, which discharges through a feed-water heater. All the steam-pumps, together with the independent air-pump, exhaust through the heater to the atmosphere.

DIMENSIONS.

Diameter of high-pressure steam-cylinder (one)	20 in.
Diameter of low-pressure steam-cylinder (one)	40 "
Diameter of plunger (one)	20 "
Diameter of each piston-rod	4 "
Stroke of steam-pistons and pump-plunger	3 ft.

GENERAL DATA.

1. Duration of trial (D)	10	hrs.
2. Boiler-pressure indicated by gauge (barometric pressure, 14.7 lbs.)	120	lbs.
3. Temperature of water in pump-well	60	degs.
4. Temperature of water supplied to boiler by main feed-pump, leaving heater	215	"
5. Temperature of water supplied by low-pressure jacket-pump	225	"
6. Temperature of water supplied by high-pressure jacket, separator, and reheater-pump, that derived from separator being 340°, and that from jackets 290°	300	"

7.	Weight of water supplied to boiler by main feed-pump	18,863	lbs.
8.	Weight of water supplied by low-pressure jacket-pump	615	"
9.	Weight of water supplied by pump for high-pressure jacket, separator, and reheater-tank, of which 210 lbs. is derived from separator....................	1,025	"
10.	Total weight of feed-water supplied from all sources	20,503	"
11.	Percentage of moisture in steam after leaving separator..	1.5%	

DATA RELATING TO WORK OF PUMP.

12.	Area of plunger minus ½ area of piston-rod (*A*)......	307.88	sq. in.
13.	Average length of stroke (L and L_s).......	3	ft.
14.	Total number of single strokes during trial (N and N_s)	24,000	
15.	Pressure by gauge on force-main (P)...............	95	lbs.
16.	Vacuum by gauge on suction-main	7.5	in.
17.	Pressure corresponding to vacuum given in preceding line (p)........................	3.69	lbs.
18.	Vertical distance between centres of two gauges.....	10	ft.
19.	Pressure equivalent to distance between two gauges (s)	4.33	lbs.
20.	Total leakage of pump during trial, determined from results of leakage-test (C)....	3,078	cu. ft.
21.	Number of double strokes of pump per minute.......	20	
22.	Mean effective pressure measured from pump-diagrams.................	105	lbs.
23.	Indicated horse-power exerted in pump-cylinders....	117.55	I.H.P.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

24.	Area of high-pressure piston minus ½ area of rod (A_{s1})	307.88	sq. in.
25.	Area of low-pressure piston minus ½ area of rod (A_{s2})	1,250.36	" "
26.	Average length of stroke, each....................	3	ft.
27.	Mean effective pressure measured from high-pressure diagrams ($M.E.P._1$)............	59.25	lbs.
28.	Mean effective pressure measured from low-pressure diagrams ($M.E.P._2$)................	13.60	"
29.	Number of double strokes per minute (line 21).......	20	
30.	Indicated horse-power developed by H.-P. cylinder..	66.33	I.H.P.
31.	Indicated horse-power developed by L.-P. cylinder..	61.82	"
32.	Indicated horse-power developed by both cylinders..	128.15	"
33.	Feed-water consumed by plant per indicated horse-power per hour, corrected for separator-water and for moisture in steam.........................	15.60	lbs.
34.	Number of heat-units consumed per indicated horse-power per hour....	15,652.1	B.T.U.
35.	Number of heat-units consumed per indicated horse-power per minute.......................	260.9	"

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER, AND COMPUTATIONS BASED THEREON.

36. Total heat of 1 lb. of steam at 120 lbs. gauge-pressure, containing 1.5% of moisture, reckoned from 0° F. = 1220.6 − (1.5% of 866.7)........................	1,207.6	B.T.U.
37. Ditto, reckoned from 215° temperature of main feed-water = 1207.6 − 215.9........................	991.7	"
38. Ditto, reckoned from 225° temperature of low-pressure jacket-water = 1207.6 − 226.1....................	981.5	"
39. Ditto, reckoned from 290° temperature of high-pressure jacket and reheater water = 1207.6 − 292.3 = ..	915.3	"
40. Heat of separator-water reckoned from 340° = 353.9 − 343.8	10.1	"
41. Heat consumed by engine (H) = (18.863 × 991.7) + (615 × 981.5) + (815 × 915.3) + (210 × 10.1) =.....	20,058,150	"

RESULTS.

Substituting these quantities in the formulæ, we have:

$$1.\ \text{Duty} = \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33}) \times \overset{L}{3} \times \overset{N}{24{,}000}}{\overset{H}{20{,}058{,}150}} \times 1{,}000{,}000$$

$$= 113{,}853{,}044 \text{ foot-pounds.}$$

$$2.\ \text{Percentage of leakage} = \frac{\overset{C}{3078 \times 144}}{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24{,}000}} \times 100 = 2.0\%.$$

$$3.\ \text{Capacity} = \frac{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24{,}000} \times 1.24675}{\overset{D}{10}}$$

$$= 2{,}763{,}716 \text{ gallons.}$$

4. Percentage of total frictions

$$= \left(1 - \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33})}{(\overset{A_{s1}}{307.88} \times \overset{M.E.P._1}{59.25}) + (\overset{A_{s2}}{1250.36} \times \overset{M.E.P._2}{13.6})}\right) \times 100$$

$$= 9.0\%.$$

In the use of a system like the preceding, every precaution should be observed in the adoption of methods, as well as in taking observations. The water discharged by a pumping-engine, for example, should never be obtained by computation from the measured dimensions of the pump and the observed number of strokes, but should be measured directly. A weir is commonly arranged for this purpose. Where the delivery of the pump has been actually measured, and the pump thus standardized, its use as a meter is less liable to error, but it is best avoided whenever possible.

Mr. J. R. Freeman has shown the practicability of making a standard orifice a substitute for a weir.* It possesses the special advantage of permitting the flow to be measured at any desired pressure and head.

189. Examples of Engine-testing, as illustrating usual current and standard practice, will better complete the discussion of this subject than any further extended descriptions of details and of methods of observation, of computation, and of preparation of reports. In a separate work by the Author are given, as fully as is possible without occupying too great space, illustrations of this character, as obtained by reference to the reports of the more expert and most experienced of contemporary engineers, or to reports of earlier work which have been regarded by engineers as best representing good practice in special departments.†

In all cases the engineer must himself judge whether to err, if at all, in making such tests and in preparing his report, in the direction of extended and complete—and hence costly—investigation and deduction, or in that of brevity and possible incompleteness. If, in any case, a doubt arises, it will usually be wisest to err on the side of completeness and accuracy.

The following table illustrates the quantities preliminarily measured and those later determined in the official trial of a naval vessel: ‡

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* Journal N. E. Water-works Assoc., 1888. Trans. Am. Soc. M. E., 1889.

† Engine and Boiler Trials; N. Y., J. Wiley & Sons.

‡ Report of Engineer-in-Chief, 1890.

FULL-POWER FORCED-DRAUGHT CONTRACT TRIALS OF TWIN-SCREW STEEL VESSELS OF THE U. S. NAVY.*

1	Name of vessel	PHILADELPHIA		SAN FRANCISCO.	
2	Date of trial.	June 25, 1890		August 27, 1890.	
3	Duration of trial (during which data were taken)	4 hours, 15 minutes		4 hours, 6 minutes.	
4	Place of trial	Off south side of Long Island and Block Island.		Santa Barbara Channel.	
5	Condition of weather	Smooth sea; light wind.		Smooth sea; light wind.	
6	Length between perpendiculars	315' 0"		310' 0"	
7	Beam	48' 6$\frac{3}{8}$"		49' 1$\frac{3}{8}$"	
8	Mean draught on trial	19' 2$\frac{1}{8}$"		18' 9"	
9	Displacement on trial, tons	4325.0		4088.0	
10	Immersed midship section on trial, square feet	815.0		770.0	
11	Coefficient of fineness, prismatic.	0.515		0.501	
12	Type of engine	Horizontal, triple expansion.		Horizontal, triple expansion.	
13	H. P. cylinder, diameter, inches.	38		42	
14	I. P. " " "	58		60	
15	L. P. " " "	86		94	
16	Stroke of piston, inches	40		36	
17	Number and type of boilers	4 double-end, 1 single-end.		4 double-end, 1 single-end.	
18	Length and diameter of each	D. E. 20' × 14', S.E. 8' 3" × 8' 6".		D. E. 19' 2" × 14' 8", S.E. 8' × 8'.	
19	Number and diameter of furnaces in each	D. E. 8, 36"; S. E. 2, 32".		D. E. 6, 42"; S. E. 1, 39".	
20	Total grate-surface, square feet, used on trial	624 main boilers only.		567.6	
21	Total heating-surface, square feet, used on trial	20458		20133.8	
22	Total condensing-surface, square feet, used on trial	13510		14518	
23	Screw-propeller, diameter	14' 6"		13' 6"	
24	Screw-propeller, mean pitch on trial	20' 4".7		18' 9"	
25	Screw-propeller, mean pitch adjustable between	18' 6" and 21' 6"		17' 9" and 19' 9"	
26	Screw-propeller, developed area, square feet	57.17		57.59	
27	Screw-propeller, number of blades	3		3	
28	Steam-pressure in boilers	159 8		135.3	
29	Air-pressure in fire-rooms or air-ducts, in inches of water	1.49		2.00	
		Starboard.	Port.	Starboard.	Port.
30	Steam-pressure at engines per gauge, pounds	151.2	154	126 2	128.4
31	Steam-pressure, first receiver, absolute, pounds	57.6	56.5	55.1	58.1
32	Steam-pressure, second receiver, absolute, pounds	24.5	24.3	19 25	18.25

33	Vacuum in condenser in inches of mercury	24.3	24.7	25.7	26.1
34	Revolutions of main engines per minute	119.58	119.55	125.8	123.83
35	H. P. cylinder, Mean pressure	61.96	57.73	51.37	46.45
36	H. P. cylinder, I. H. P.	1645.1	1560.47	1590.06	1427.98
37	I. P. cylinder, Mean pressure	21.3	19.91	25.34	26.8
38	I. P. cylinder, I. H. P.	1352.17	1325.24	1603.63	1700.43
39	L. P. cylinder, Mean pressure	8.91	9.99	9.92	10.64
40	L. P. cylinder, I. H. P.	1249.73	1400.46	1594.39	1664.29
41	Aggregate equivalent mean pressure on L. P. piston	30.34	30.63	30.32	30.68
42	Collective I. H. P. each main engine	4247.01	4286.17	4788.08	4792.64
43	Collective I. H. P. both main engines	8533.18		9580.72	
44	I. H. P. air-pumps	31.56	33.32	66.48	41.22
45	I. H. P. circulating-pumps	30.27	24.69	14.36	15.07
46	I. H. P. feed-pumps	34.48		42.85	
47	I. H. P. blowers	87.59		110.67	
48	I. H. P., other auxiliaries	39.70		41.55	
49	Aggregate mean I. H. P. of machinery	8814.79		9912.93	
50	Aggregate maximum I. H. P. of machinery	9021.47		10604.32	
51	Speed per hour in knots	19.68		†19.52	
52	Slip of propeller, per cent	18.43	18.49	16.11	14.80
53	Indicated thrust of main engines only, per square foot of developed area of screw	1005.4	1015	1163.3	1182.8
54	I. H. P. per square foot of grate, based on mean I. H. P.	14.13		17.46	
55	Heating-surface per I. H. P., based on mean I. H. P.	2.32		2.03	
56	Condensing-surface per I. H. P., based on mean I. H. P.	1.53		1.46	
57	Weight of propelling machinery, tons, including water	851.62		914.12	
58	I. H. P. per ton of machinery, mean I. H. P.	10.35		10.84	
59	I. H. P. per ton of machinery, maximum I. H. P.	10.59		11.60	
60	Total coal burned per hour, pounds	25818.0		....	
61	Refuse from coal per hour, pounds	1678.0		....	
62	Coal per hour per square foot grate-surface	39.35		....	
63	Coal per hour per I. H. P.	2.93		....	

* Mr. Froude finds that, under the conditions of operation usual in successful vessels, the best area of screw-disk is, in square feet,

$$A = 9R/V^2, \text{ nearly;}$$

the resistance, R, being given in pounds, and the speed, V, in knots. The angle at which the blade should stand at the assumed radius, to secure maximum efficiency, is 45° with its path.

† Speed for last half-hour between points on shore 20.17 knots; mean speed for trial by patent log 20.6 knots.

A trial of a small compound engine made under the direction of the Council of the Society of Arts, to determine its valve as a motor for electric-lighting purposes, gave the following data.* The engine was of 5″.24 and 8″.98 diameter of cylinder, and of 14″ stroke of piston.

The data obtained from the log of the second trial being plotted, gave a graphical representation of the whole course of the trial, thus:

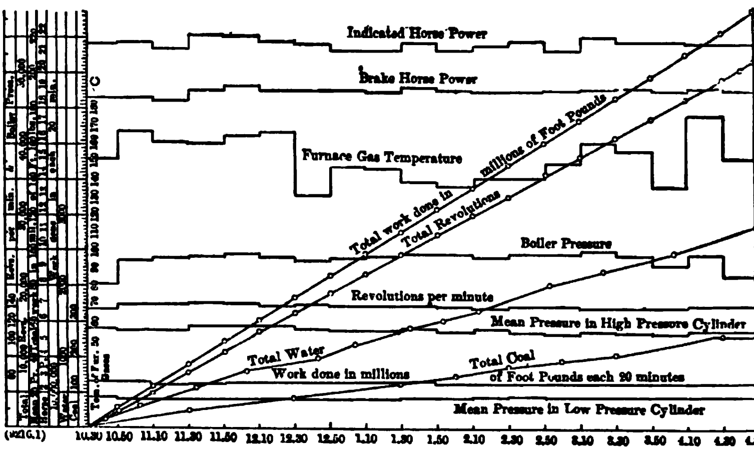

FIG. 199.—GRAPHICAL LOG.

Engine.—The work done, 21.55 indicated h. p., corresponds to 55,300 thermal units per hour, or 12.0 per cent of the whole heat taken up by the water. The efficiency of a perfect heat-engine working between 383° and 212° F. would be 0.205. Such an engine receiving the same amount of heat as the Paxman engine, namely, 460,300 thermal units per hour, would turn into work 93,200 thermal units per hour. The actual efficiency of the engine therefore, compared with such a perfect engine, is 59 per cent. The heat received by the engine per indicated h. p. per hour was 21,350 thermal units. The brake h. p. of the engine was 18.95; its mechanical efficiency was

* Journal Society of Arts, Feb. 15, 1889, p. 235.

TEST OF A COMPOUND ENGINE.

1	Date	Sept. 26	Sept. 28	Oct. 1	Oct. 1
2	Trial	A1	A2	B	C
3	Duration	6.43 hours	6.27 hours	3.12 hours	¼ hour
4	Power	Full	Full	Half	Empty
5	Revolutions per minute	140.48	137.35	138.10	144.20
6	Mean initial pressure, high-pressure cylinder	176.6	176.8	113.0	....
7	Mean ratio of expansion				
8	Mean effective pressure, high-pressure cylinder	52.93	54.98	33.25	11.32
9	Mean effective pressure, low-pressure cylinder	17.56	16.78	8.92	−.19
10	Indicated H. P. effective pressure, high-pressure cylinder	11.14	11.30	6.85	+2.40
11	Indicated H. P. effective pressure, low-pressure cylinder	10.98	10.26	5.47	−0.12
12	Indicated H. P., total	22.12	21.56	12.32	2.28
13	Brake-load, net	288.8	288.0	147.6	....
14	Brake H. P	19.44	18.95	9.76	....
15	Mechanical efficiency	0.879	0.879	0.792	...
16	Indicated H. P. in driving engine	2.68	2.61	2.56	2.28
	Boiler.				
17	Mean boiler-pressure (above atmosphere)	191.35	187.98	120.10	....
18	Atmospheric pressure for the day	14.9	14.8	14.8	....
19	Boiler-pressure (absolute)	206.25	202.78	134.9	....
20	Temperature of boiler-steam	384.3°	382.9°	350.1°	....
21	Pounds of feed used per hour	[448.7]	447.1	392.2	....
22	" " " indicated H. P. per hour	[20.28]	20.74	26.72	....
23	" " " brake H. P. per hour	[23.08]	23.59	33.73	....
24	Mean temperature of feed-water in tank	63.0	63.0	69.9	....
25	" " " before entering coil	[201]	201	....	....
26	Mean temperature of feed-water before entering exhaust-feed heater	115.7	63.0	..	....
27	Temperature of chimney gases (Fahr.)	355.4°	304.4°	369.1°	....
28	Coal per hour	39.66	40.70	27.25	....
29	" " " per indicated H. P	1.79	1.89	2.21	....
30	" " " per brake H. P	2.04	2.15	2.79	....
31	Pounds of water per pound of coal	....	10.99	12.08	....
32	" " " " from and at 212° F	....	11.71	12.76	....

DISTRIBUTION OF HEAT.

	Thermal Units.	Percentages.
Caloric value of 40.7 pounds of coal	577,900	100
Heat expended in heating and evaporating water, including heat given up by gases to coil in chimney	460,300	79 65
Heat expended in raising temperature of furnace-gases	40,700	7.05
Heat lost by radiation	51,070	8.85
Heat lost by imperfect combustion	15,450	2.68
Heat expended in evaporating moisture in coal	570	0.10
Heat lost in ash and otherwise unaccounted for	9,810	1.67
	577,900	100.00

therefore 87.9 per cent, the indicated h. p. expended in driving the engine itself being 2.61.

Boiler and Engine.—The combined efficiency of the furnace, boiler, and engine, as represented by the consumption of fuel per horse-power, works out to 9.6 per cent, 55,300 thermal units being turned into work per hour, with an expenditure of fuel having a value of 577,900 thermal units. The coal used per indicated h. p. per hour was 1.89 pounds, and per brake h. p. per hour 2.15 pounds.

Steam per Indicator-cards.—The amount of steam shown by the indicator-diagrams was as below:

	Percentage of whole weighed feed-water.
Steam in h. p. cylinder at a pressure of 150 lbs. per square inch above the atmosphere, which corresponds to a point at 0.39 of the stroke, a little after cut-off in all cases.......	65.0
Steam in l. p. cylinder at a pressure of 10 lbs. per square inch above the atmosphere, which corresponds to a point at 0.67 of the stroke, well before release in all cases	78.8

The table on page 665, exhibiting the distribution of heat in a small engine of the Corliss type, as given by Professor Peabody, illustrates also the extraordinary extent to which wastes may take place in an engine of small power.*

This engine was 8 inches diameter of cylinder and 24 inches stroke of piston. Large non-condensing engines of this type ordinarily demand 30 pounds of steam per horse-power per hour or less.

190. The Indicator and the Dynamometer are the instruments employed in the engine-test proper. The purpose of their use is the measurement by the one of all fluctuations of pressure and of volume of the steam within the working cylinder, and of the work done and power developed by its action on the piston, the gross work performed by the transformation of heat-energy; and by the other the net work of the engine, the work done and power available at the engine-shaft for useful application. The difference between these two

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* Peabody's Thermodynamics.

TESTS OF HARRIS-CORLISS ENGINE, MASSACHUSETTS INSTITUTE OF TECHNOLOGY.

No. of test.	Boiler-pressure above the atmosphere, pounds.	Horse-power of the engine.	Revolutions per minute.	Crank end. Per cent cut-off.	Crank end. Per cent release.	Crank end. Per cent compression.	Crank end. M. E. P.	Crank end. Horse-power.	Head end. Per cent cut-off.	Head end. Per cent release.	Head end. Per cent compression.	Head end. M. E. P.	Head end. Horse-power.
309	70.9	2.916	60.87	2.5	97.0	1.5	5.63	1.014	4.0	97.0	2 5	10.21	1.902
310	71.0	3.905	61.10	5.0	97.0	2.0	10.17	1.839	5.5	97.0	3.5	11.05	2.066
301	71.1	5.502	59.31	7.0	95.5	2.5	13.53	2.375	8.5	95.5	5.0	17.23	3.127
300	70.8	5.627	59.74	7.5	95.5	2 0	14.13	2.499	9.0	95.5	4 0	17.11	3.128
308	71.7	6.118	61.36	18.0	96.0	1.0	17.38	3.157	15.0	96.0	2.0	15.74	2.956
311	73.8	6.210	61.45	18 0	97.0	1.0	17.02	3.096	20.0	97.0	3.0	16.56	3.114
304	72.7	6.896	60.43	11.0	96.0	1.7	17.82	3.188	11.0	96.0	2.5	20.05	3.708
306	71.9	7.416	60.54	11.0	97 0	2.0	18.45	3.310	13.5	97.0	3.5	22.16	4.106
305	72.2	7.586	60.03	10.0	95 0	2.5	19.13	3.399	14.5	96.0	4.0	22.52	4.137
312	72.0	8.105	61 20	20.5	98.0	1.0	20.97	3.800	21.5	98.0	2.0	22.98	4.305
307	76.5	9.586	60.67	18.5	96.5	1.0	25.61	4.599	17.5	97.0	2.0	26.86	4 987
303	72.0	10.084	59.44	20.0	95.0	1.5	27.58	4.852	20.0	96.0	2 0	28.49	5.182
313	69.2	10.806	60.80	13.0	97 0	1.0	20.64	3.714	28.0	97.0	1.6	35.43	6.592
302	71.9	11.694	60.36	19.0	95.0	2.0	26.92	4.810	29.0	96.0	3.0	37.27	6.884

No. of test.	Water per H. P. per hour. By tank.	Water per H. P. per hour. By indicator.	Weight of steam per stroke. Crank end. Cut-off.	Crank end. Release.	Crank end. Compression.	Head end. Cut-off.	Head end. Release.	Head end. Compression.	Weight of mixture per revolution.	Per cent of mixture accounted for by indicator as steam. Cut-off.	Per cent of mixture accounted for by indicator as steam. Release.	Diff.	Lbs. re-evaporation per H. P. per hour.
309	78.14	54.10	.0075	.0205	.0016	.0111	.0269	.0026	.0626	29.7	75.7	46.0	36.1
310	57.94	42.90	.0102	.0230	.0015	.0115	.0273	.0028	.0665	32.6	75.6	43.0	26.6
301	48.07	27.02	.0127	.0256	.0018	.0170	.0230	.0032	.0811	36 6	71.0	34.4	18.1
300	48.08	33.76	.0130	.0256	.0018	.0175	.0325	.0034	.0807	37.9	72.1	34.2	17.2
308	46.42	33.90	.0184	.0287	.0014	.0178	.0312	.0022	.0806	44.9	74.3	29 4	14.3
311	45 06	33.10	.0183	.0275	.0015	.0200	.0323	.0025	.0799	47.9	74 9	27 0	12.8
304	42.16	29.70	.0176	.0271	.0016	.0197	.0335	.0024	.0842	44.3	72 0	27.7	12.2
306	42.22	29.90	.0174	.0280	.0017	.0219	.0371	.0024	.0903	43.5	72.1	24.6	12.6
305	40 46	28.40	.0162	.0311	.0019	.0233	.0329	.0027	.0893	44.2	71.7	27.5	11.7
312	42.20	31.10	.0219	.0326	.0013	.0256	.0395	.0022	.0966	49.2	74.6	25 4	11.1
307	38.07	25.90	.0247	.0336	.0012	.0272	.0381	.0022	.1036	50.1	69 2	19.1	7.5
303	37.51	25.30	.0273	.0351	.0015	.0284	.0399	.0024	.1094	50 9	68.6	17.7	6.9
313	35.69	27.40	.0195	.0296	.0013	.0400	.0512	.0021	.1040	57.2	77.7	20.5	7.5
302	37.08	25.30	.0261	.0351	.0019	.0405	.0511	.0028	.1243	53.6	69.4	15.8	6.1

quantities is the measure of the lost energy and the wasted power, due to the resistances of the machine itself, the sum of the friction-resistances and the back-pressure on the exhaust side of the piston, if the gross indicated power is measured to the line of external atmospheric or condenser pressure, or to friction alone if the power is taken as exclusive of back-pressure work.

The dynamometer is sometimes of the transmitting form, stationed between the engine and its work or any introduced resistance; but it is most usually of the type known as the Prony brake or the absorbing dynamometer, and takes up the whole external power of the engine, converting all that energy into heat; which heat it wastes by conduction and radiation to surrounding objects or to a stream of water kept flowing over it or through the rim of the brake-wheel.

191. The Principles of the Indicator of the usual construction and type, those which govern its action and determine its value, are as follows:

(1) It must exhibit with precision the pressure of the steam within the working cylinder at every instant throughout the stroke.

(2) Simultaneous measures must be given of the position of the piston corresponding to the given pressure, each instant.

(3) The diagram produced must be so made, automatically, as to have its ordinates exactly proportional to the steam-pressures and its abscissas as accurately proportional to the motions of the piston, each point in the curve, by its co-ordinates, giving a measure, simultaneously, of these two quantities.

(4) The diagram must be unaffected either by the forces acting on the engine, other than that which it is constructed to measure, or those brought into existence by its own motions, and whether they are active or passive, whether of inertia or of friction. The ideal indicator would be an instrument possessing the above qualities, and would trace a conveniently large diagram with absolute exactness. It would be free from inertia, and perfectly inflexible in every part. As these ideal

conditions are approximated, differences among the best makes of indicators become less and less, and should finally disappear. As they are now sometimes made, however, unless carefully selected and as carefully tested and standardized, it is perfectly possible for differences of very considerable importance to be observed.

192. The Essentials in Construction of the Indicator are:

(1) Such form and construction as will insure its meeting the prescribed general conditions—accuracy of representation of the variations of steam-pressure and the simultaneous movement of the piston at all times.

(2) Such simplicity of form as will make it free from liability to accident and failure in operation.

(3) Such lightness of parts, and such rigidity as a whole, as will prevent any inaccuracy of indications arising from its inertia.

(4) It should be easily, conveniently, and safely attachable and removable, and readily and handily manipulated.

Stiffness, lightness, and exactness of standardization are the prime essentials. The springs should be exactly standard; the moving parts as light as is consistent with proper strength and stiffness; the stationary parts should be carefully proportioned and rigid; the whole instrument should be portable, and yet the scale of its diagram as large as practicable, and consistent with exactness in its production.

193. The Forms of the Indicator, as commonly constructed, are usually very similar, the more important differences being found in the recording system. The original indicator employed from about 1814* by Watt consisted of a small steam-cylinder traversed by a piston, the latter held by a spring, which was compressed or extended proportionally to the pressure, the cylinder being placed in communication with the interior of the working cylinder by a pipe of sufficient size, and fitted with a cock by means of which the steam could be

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*See Tredgold on the Steam-engine. London, 1827.

cut off from the instrument at any instant. So long as this cock was open, the indicator, if properly mounted, and the main steam-piston were affected by precisely the same intensity of pressure, and the movement of the piston of the former was a measure of the pressure. A pencil was attached to the indicator-piston, and its point recorded all such variations of

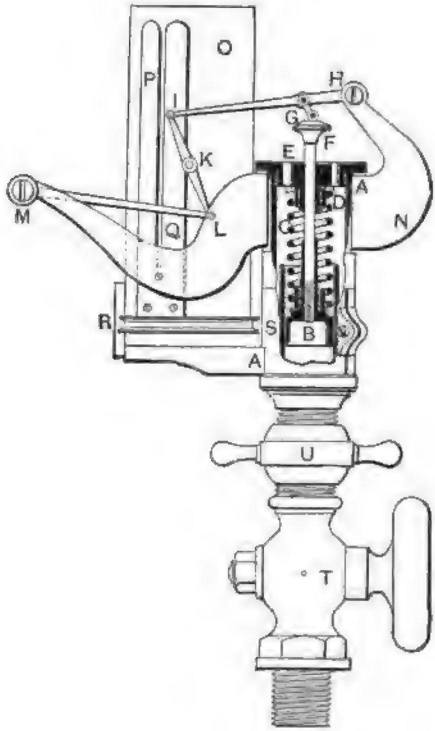

FIG. 200.—THE RICHARDS INDICATOR.

steam-pressure on a movable slate, which was so connected with the mechanism of the engine as to move in exact coincidence with the main piston, and precisely at right angles to the line of motion of the pencil. Thus the abscissas of the curve produced were proportional to the motions of the piston, and the ordinates of the same points in the curve gave the simultaneous pressures.

In the later instruments of McNaught and Hopkinson, metal cylinders revolving on vertical axes were substituted for the sliding panel of Watt's arrangement, and a much more compact instrument was thus made.

The later devices have been introduced with a view to securing lightness of parts and reduced motion.

Fig. 200 is a sketch, partly in section, of the first of the later type of instrument, the Richards indicator, invented by Professor C. B. Richards about 1860. *AA* is the cylinder; *B* is the piston, connected by a properly made spring *CD* with the cap *E* of the barrel. The head of the piston-rod *F* is attached by a link *G* to the lever *HI*, by means of which a comparatively large motion of the pencil *K* is obtained without much movement of the piston and its attached parts, and consequently with but little inertia-effect. A parallel motion of the Watt type, *HI*, *KLM*, guides the pencil-holder *K* in a right line parallel to this path of the piston of the indicator. The paper is wrapped about the cylinder *O* and secured at its ends by the clamps *PQ*. The paper cylinder is turned on its axis by a cord on the pulley *RS*, which cord is attached to some form of "reducing motion" which causes it to move with the engine-piston.

Communication with the engine-cylinder is established by a steam-passage through the cock *T*, and the instrument is secured in place by the clamp *U*. When in action, this cock is opened; the indicator-piston rises and falls with the varying pressure in that end of the engine-cylinder, and the paper barrel rotates backward and forward as the engine-piston moves. When all is ready, the instrument being heated up and working smoothly, the pencil is pushed lightly against the paper, and a diagram is drawn, representing all changes of pressure and volume of the working fluid during the period of contact. This modification of the indicator was found to give satisfactory results up to a comparatively high speed, and its limit of efficiency was determined by the degree to which the lightening of its parts could be safely carried.

A still later form (1875) is that of Mr. J. W. Thompson,

Fig. 201. In this indicator the same general style is retained, but the parallel motion is modified. The cylinder AA contains a piston, B, connected by a spring, as before, to the cap DE; while the head F of the rod actuates a pencil-arm HK, and a parallel motion is obtained by linking on LI from the standard LM, and G from the swivelling support MN, which also carries L. The action of the instrument when in use is precisely as before; its decreased weights of moving parts, however, enabled it to be confidently relied upon at speeds far above those of even the Richards instrument. The old McNaught indicator became unsatisfactory at about 60 revolutions per minute; the Richards carried this limit well up toward and sometimes above 200 revolutions; while the Thompson indicator was found capable of doing good work on even the fast engines of the most modern type at the date of its invention. The most recent and a still lighter style of this instrument is shown in Fig. 202.

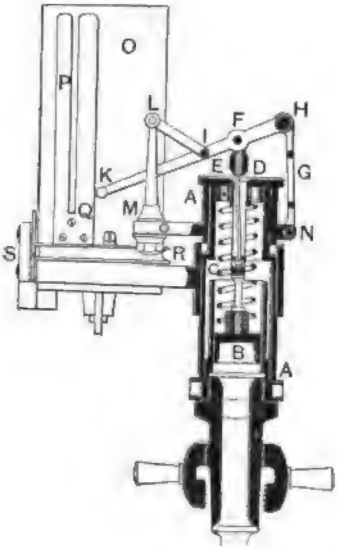

FIG. 201.—THE THOMPSON INDICATOR.

The later improvements consist in lightening the moving parts, substituting steel screws in place of taper pins, using a light steel link instead of a brass one, reducing the weight at the pencil-lever and elsewhere, shortening the length and reducing the weight of the paper cylinder one half, and reducing friction to a minimum.

The paper cylinder is so constructed that the tension of the coiled drum-spring within it can be varied for different speeds. As little or as much of the spring can be taken up or let out as desired, thus providing for fine adjustments.

Sufficient tension should be given to keep the cord taut at all points. When exceptionally accurate work is desired, the

length of the diagram may be carefully measured, and compared with the length of a line traced on the paper when the engine is moved slowly. If the diagram is found to differ in

FIG. 202.—THE THOMPSON INDICATOR.

length from this line, vary the tension of the spring till they agree.

All these indicators are provided with a piston 0.798 inch diameter, ½ inch area, and with springs for indicating pressures up to 250 pounds. When higher pressure is to be indicated, an extra piston 0.564 inch diameter, ¼ inch area, is used, which, when substituted for the other piston, doubles the capacity of each spring, thereby adapting the indicator for indicating pressures up to 500 lbs.

The Tabor Indicator, Fig. 203, is the invention of Mr. H. Tabor (1879). In this indicator a stationary plate containing a curved slot is firmly secured in an upright position to the cover of the steam-cylinder. This slot serves as a guide and controls the motion of the pencil-bar. The side of the pencil-bar carries a roller which turns on a pin, and this is fitted so as to roll freely from end to end of the slot, with little lost mo-

tion. The curve of the slot is so adjusted and the pin attached to such a point that the end of the pencil-bar, which carries the pencil, moves up and down in a straight line when the roller is moved from one end of the slot to the other. The curve of the slot just compensates the tendency of the pencil-point to move in a circular arc, and a straight-line motion results. The outside of the curve is nearly a true circle, with a radius of one inch.* The steam-cylinder and the base of the paper-drum are made in one casting. Inside the steam-cylinder is a mov-

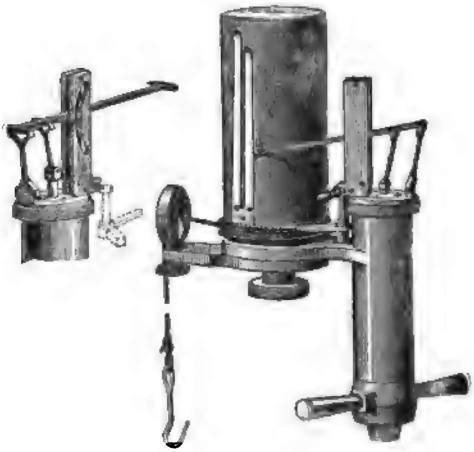

FIG. 203.—THE TABOR INDICATOR.

able lining-cylinder, within which the piston of the indicator works. This cylinder is attached by means of a screw-thread at the bottom, and openings at the opposite sides at the top are provided for the introduction of a tool for screwing it in or out. Openings through the sides of the outer cylinder are provided to allow the steam which leaks by the piston to escape. The pencil mechanism is carried by the cover of the outside cylinder. The cover proper is stationary, but a nicely fitted swivel-plate, which extends over nearly the whole of the cover, is provided, and to this plate the direct attachment of the pencil mechanism is made. By means of the swivel-plate

* "The Tabor Indicator," G. H. Barrus; N. Y. 1888.

the pencil mechanism may be turned so as to bring the pencil into contact with the paper-drum, as is done in the act of taking a diagram.

The pencil mechanism is attached to the swivel by means of the vertical plate containing the slot, which has been referred to, and a small standard placed on the opposite side of the swivel for connecting the back link. The connection be-

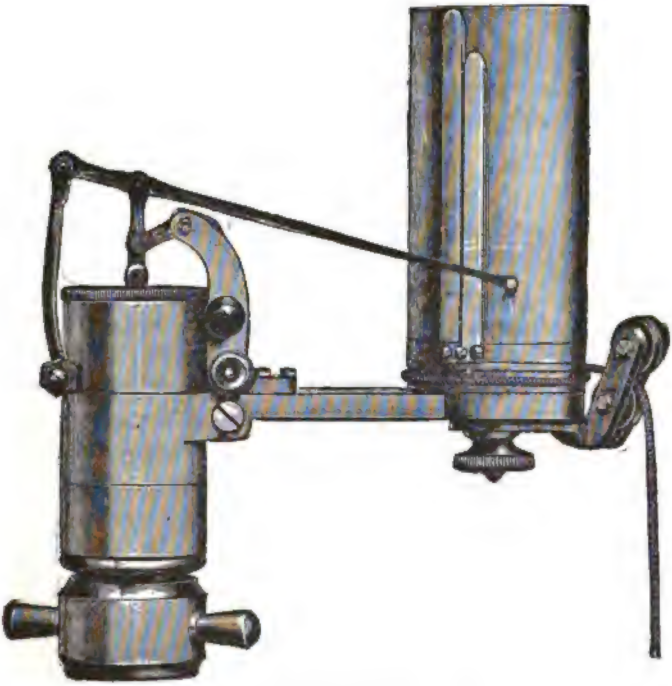

FIG. 204.—THE CROSBY INDICATOR.

tween the piston and the pencil mechanism is made by means of a steel piston-rod. At the upper end where it passes through the cover it is hollow, and has an outside diameter measuring $\frac{8}{16}$ of an inch. At the lower end it is solid, and its diameter is reduced. It connects with the piston through a ball-and-socket joint. A number of shallow grooves are cut upon the outside of the piston to serve as a so-called water-packing,

The springs used in the Tabor Indicator are of the duplex type, being made of two spiral coils of wire with fittings, as shown in the cut. The springs are so mounted that the points of connection of the two coils lie on opposite sides of the fitting.

The Crosby Indicator, Fig. 204, is still another successful recent type of the instrument (1879). In this case a still different form of parallel motion, light, stiff, and carefully adjusted, guides a very light pencil-holder carried at the end of a correspondingly light steel arm. The general arrangement of the indicator-barrel and the paper cylinder, with their attachments, is quite similar to those observed in the Richards and its successors.

When the cord has the maximum other resistances to overcome, the drum-spring should offer minimum resistance. At the beginning of the stroke, when the spring is overcoming the inertia and friction of the drum, its resistance should be a maximum, and should gradually decrease. Here a short spiral drum-spring is adopted, giving at the beginning of the stroke a comparatively slight resistance, which gradually

194. The Standardization and Test of the Indicator should always precede its use. To give satisfactory results, the instrument must give a diagram of which the abscissas shall exactly represent the successive positions of the moving piston, while the ordinate of each as exactly measures the simultaneously occurring pressure within the working cylinder. The weight and inertia of every moving part of the indicator introduce errors which, while they may not be completely eliminated, may, by reduction of size and special expedients, at least be rendered so small as to be unimportant at any ordinary speeds. They are, however, more difficult of prevention as the speed of rotation and the pressures adopted increase. In a well-made instrument the spring will precisely measure pressures, the pressure in the indicator will be sensibly the same as in the engine, and its piston will move so freely as not to affect the indications by its resistances due to close fitting. In all, however well made, on the other hand, it is found impossible, by any art of design or construction, to wholly eliminate the in-

fluences of inertia of moving parts, friction of joints and guides if the latter be used, and of piston or pencil, or the effects of variation of spring tension on the motion of paper cylinder and card. Standardization is the process of detecting and measuring such errors as are observable in the action of the apparatus, and the determination of their influence on the indications obtained by its use.

Springs are tested most satisfactorily by connecting the indicator, with its appropriate spring in place, with a small steam-boiler or steam-reservoir or convenient steam-pipe, in such manner that simultaneous measurements of pressure may be taken by the indicator record and a standard test-gauge known to be correct. If the spring be tested cold, it will be found inaccurate if it had been found right when hot; and the correct reading of a cold spring is evidence that it is not right under steam;* since, when the indicator is in use and the spring heated, both by the steam leaking past the piston and by conduction through its attachments from the piston and from the indicator-barrel, its strength and its elasticity are sensibly modified, the spring being thus weakened. In thus testing springs, such arrangements should be made as will enable the observer to hold the pressure at any desired point until readings from the standard test-gauge can be deliberately and precisely taken. If the spring is found unreliable, it should be at once exchanged for a good one.

"Throttling," or loss of pressure between the engine and the indicator, is produced by long, tortuous, or contracted passages. The connections should be as large, as straight, and as short as practicable, and, other things being equal, that indicator is best in which steam connections can be best effected. The cock under the indicator should have an opening fully as large as the pipe itself. It should also have a hole bored in from one side for the purpose of freeing the instrument and connections from water of condensation coming over with the steam. The holes in the cock or in the upper part of the

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* This difference has been found to amount to 2½ or 3 per cent.—Proceedings of Brit. Inst. C. E., 1885, vol. LXXXIII, Brightmore; discussion.

barrel of the indicator should be large enough to permit free egress to any steam that may leak past the piston; and the latter is, in all good makes, so loose as to leak observably under pressure. The effect of leakage is insensible; but were the fit a tight one, the resulting friction might be important. It is to insure this exact correspondence of pressure in the working-cylinder of the engine and in that of the indicator that it is customary, on all high-speed engines, to employ indicators simultaneously at both ends, thus obtaining very short and direct steam-connections.

The springs should be so made and fitted that their action under pressure, and when in use, may not throw the piston out of line or cramp it in the barrel, and thus produce what are sometimes found to be serious errors. A small leak does no harm, and is, on the whole, desirable. The piston should move so freely that, the spring being removed, the breath may blow it from end to end of its barrel and draw it back again.

The friction of the pencil on the paper is probably closely proportional to the force with which it is pressed upon the latter, and variable with the texture of the paper and the sharpness of the pencil-point, and its material. This is often an important source of error in the diagram. The reacting effect of this pressure on the pencil mechanism is also, but in comparatively slight degree, a source of inaccuracy of record. The result of such frictions is the production of an enlargement of the "cord" to the extent often of a very appreciable and sometimes of an important amount. This friction is sometimes relied upon to diminish those oscillations of the instrument which at high speeds render the diagram difficult of measurement, or even untranslatable. In such cases the real power of the engine may be several per cent less than shown by the instrument. To avoid this difficulty, it is best to make the pencil bear on the paper only just hard enough to make a visible mark. A hard lead-pencil or one of soft metal smoothly pointed, paper having a "metallic" or glazed surface, and a light, steady pressure producing an extremely fine but perfectly visible line are the conditions to be sought. In such case the error due to friction of pencil will be inappreciable.

The stretching of the cord turning the paper barrel is a common cause of inaccuracy in length and of distortion of the diagram. The varying tension of the spring, and the surges due to the inertia of the rotating mass, together cause variations of length of the cord that may give rise to errors of really important magnitude. The string, even when its primitive stretch is taken out of it by a preliminary application of a heavy load, retains some elasticity, and will have a sensibly variable length under the constantly varying pull when in use. Any observable friction of the paper barrel also tends to exaggerate this action. The inertia of the drum tends to compensate this action, and it is possible to so adjust the strength of the spring to this inertia as to make the variation of stress on the cord comparatively small.

The effect of this stretch is to cut off a part of the diagram at one end, and it is perfectly possible thus to reduce the apparent indicated power of the engine 10 or even 20 per cent below the correct quantity.* The longer the cord, the greater the error; and differences of sensible amount may often be detected between the diagrams from opposite ends of the same cylinder, in area of diagram and in point of cut-off and amount of expansion, produced by differences in length of the cord used. The higher the speed of rotation of the engine, the greater the amount of this error; and instruments giving perfectly satisfactory cards at low speed may produce very defective diagrams at high speeds. The lighter the drum and the spring found practicable, the better the results. A cord should always be well stretched before use; but a fine steel wire is much to be preferred. The paper cylinder or drum should have carefully adjusted springs for fast work. The difference in its initial and final tension should be as nearly as possible equal to the inertia-stresses of the drum. Improvement has been carried so far in the reduction of weight of paper barrel as to bring it, in one case at least, as low as 209 grammes (3088 grains). It has been often proposed to use aluminium for moving parts in order to reduce inertia-effects to

* Ibid.

a mininum. Errors due to this action, in good instruments, fall much below 1 per cent. In adjusting the instrument, the tension of the drum-spring should vary as the square of the number of revolutions; and it should be set for any speed in such manner that the length of the diagram should be the same at starting as when at full speed, as nearly as possible.

Oscillations of the pencil about its proper position, and the consequent production of wavy lines in the diagram, cause the most serious defects in diagrams taken at high speeds of engine. Such deformations of the diagram are due to the inertia of the pencil and its holder and connection, and become greater as speeds increase, until, with every instrument, a speed is finally reached at which the diagram becomes unintelligible, as in the accompanying figure, which represents the card obtained by using an old style of indicator, with a light spring, at 300 revolutions per minute. With the best modern indicators it is easy to secure a perfectly smooth diagram at this speed.

The best indicator, as is now evident, is that which, by such comparison and examination as has been described, is found to give the most exact and reliable diagram, and to be least affected by inertia-forces and the action of its own parts at high speed; it is that, in detail, which has proportionally the largest and lightest piston, the stiffest and lightest springs, the least friction of moving parts, the most perfect pencil mechanism, the most accurate and constant scale of pressures, the most perfect adjustment of drum-spring, and the lightest moving parts generally.

195. **The Attachment of the Indicator** should always be so effected that its piston may receive precisely the pressure simultaneously acting on the engine-piston, and so that the motion of the paper shall exactly reproduce, as to time and in its proper proportion, the movement of the piston. This means that the steam-connection should be amply large and free from bends and angles, and that the cord and reducing motion giving movement to the paper cylinder or drum should be so arranged as to lead right, and to be perfectly free from lost motion or stretch.

In attaching the instrument, it is usual to drill a half-inch

hole in each end of the steam-cylinder, and to make connections with half-inch pipe to the indicator-cock as directly as possible. In many cases it will be found that the drilling has already been done by the builder of the engine. The opening into the cylinder is commonly in the clearance-space back of the piston. Care should be taken that it is not covered by the piston at the end of stroke, and that the in-rush of steam from the steam-port is not likely to produce any sensible effect by blowing across the hole. Especial care should be taken to prevent chips from the drill falling into the cylinder and lodging where they can do injury. The work should, if practicable, be done with the heads removed; if this is not practicable, a little steam should be turned on and the chips blown out before starting. If the indicator-cock can be screwed directly into the cylinder, it is an advantage. The indicator should, if possible, stand in the vertical position when in use, and one should be placed at each end of the cylinder, and diagrams taken as nearly simultaneously as possible. The cock between the indicator and the cylinder should be of the full size of the pipe, and should be so made that steam may be at any time either turned on the instrument or blown out into the air to clear the passages, and to see that all is right.

The Reducing Motion is made in many ways, and is often, by the ingenious engineer, improvised for the occasion. It must reduce the motion of the piston so as to give a correct throw at the drum and exactly proportionally at every part of the movement. Such apparatus is fully described in all the many well-known works especially devoted to the subject, and need not be here further considered.*

196. Indicator-diagrams, taken under proper conditions and with good instruments, are diagrams of energy on which the ordinates measure the varying pressures in the cylinder, corresponding to the positions of the piston as measured off by the simultaneous abscissas of the diagram; while the area

* For a description of a variety of the best, see "Handbook of Engine and Boiler Trials," chap. IV.

represents the work done by the steam on the piston of the engine. The forms and relations of the several lines of which the diagram is composed reveal the method of action of the valves and the effectiveness of the pipes and passages as conduits for the entering and the exhausted steam. The correct interpretation of the diagram thus becomes an exceedingly important matter.

197. The Typical Diagram and its Nomenclature, assuming the indicator applied to the steam-engine, are as below.

The curves described on the indicator-cards of engines present many differences as to the mode in which pressure and volume vary, and their figures cannot be expressed by any mathematical formulæ; since it is impossible to separate those irregularities which arise from fluctuations in the pressure of the steam from those which arise from the friction and inertia of the moving parts of the indicator, and also because the law of such changes as actually take place in the cylinder of the engine is not precisely known.

An approximate form of diagram is therefore taken in theoretical treatment, which is that which is approached more and more closely as the machine is improved. Fig. 205 is such a diagram. AB represents the volume of the mass of steam when admitted into the cylinder. The *first assumption* is that the pressure of the steam remains constant during admission, so that AB is a line parallel to OX, and the pressure is represented by $OA = GB$. The *second assumption* consists in assigning to the curve BC one or other of two definite forms:

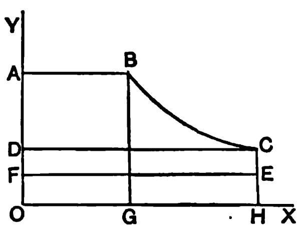

FIG. 205.—IDEAL DIAGRAM.

(1) When the cylinder has no steam-jacket, the steam is assumed to expand without receiving or giving out heat; so that BC is an *adiabatic curve.*

(2) When there is a steam-jacket, it is assumed that the heat communicated by means of that jacket is just sufficient to prevent any appreciable part of the steam from becoming

liquid; so that BC is a curve of pressures and volumes of saturated steam.

The indicator-diagram, although generally assumed to represent the variation of the effort on the piston of the engine at each half-revolution, really exhibits only one part of that action at any given instant. The line $ABCF$, Fig. 206, exhibits the effort of the steam during the forward stroke; but that effort is partly equilibrated by the back-pressure and the compression on the opposite side. If these are represented by the line

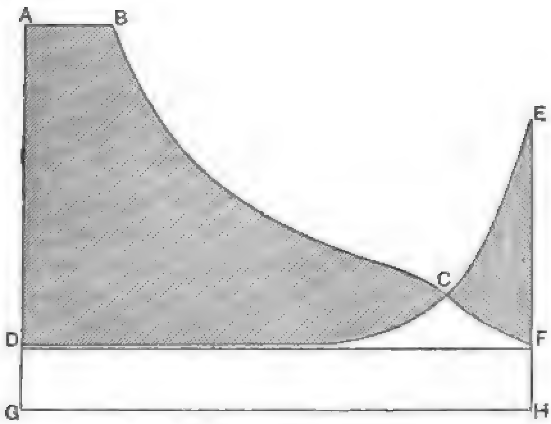

FIG. 206.—EFFORTS ON THE PISTON.

DCE, it is evident that the real variations of net effort are exhibited by the space $ABCD$ and by CEF; the former being positive, the latter negative. It is thus necessary to combine parts of two opposite simultaneous diagrams to ascertain the real pressures transmitted from the piston.

198. The Causes of Modified Forms of Diagram are usually simple, and easily traced. The actual form of the diagram differs from the ideal form, as just described, in consequence of the occurrence of a number of conditions which are usually more or less objectionable. These conditions are classed thus:

Causes which affect the power of the engine, as well as the figure of the diagram:

(1) Wire-drawing in taking steam and at cut-off.
(2) Clearance in the cylinder and passages.
(3) Compression, or cushioning.
(4) Prerelease.
(5) Conduction of heat by the metal of the cylinder.
(6) Liquid water present in the cylinder.

Causes which affect the figure of the diagram only:

(7) Undulations in the motion of the pencil.
(8) Friction of the indicator.
(9) Position of the indicator.

199. **The Interpretation of Diagrams** is usually easily effected, and by means of this "engineer's stethoscope" it becomes possible to ascertain the nature and cause of almost every defect in the distribution of pressures and volumes of the working fluid, as in the adjustment of the valve-motion, and the size or proportions of steam-passages, or of the connecting pipes. The power exerted by the steam is easily measurable. These several points may be summarized thus:

(1) Gross power exerted by the steam.
(2) Net power of the steam, and equivalent net power of the engine.
(3) Resistance of unloaded engine.
(4) Net power of the engine.
(5) Details of various wastes of power, as by wire-drawing, back-pressure, etc.
(6) Valve-adjustments.
(7) Effectiveness of valve-gearing.
(8) Adequacy of sizes of port.
(9) Quantity of steam present at any point in the stroke.
(10) Feed-water demanded, exclusive of that wasted by cylinder condensation.
(11) With a boiler-trial, the actual expenditure of steam, fuel, and money, for a given amount of power; and wastes by leakage and condensation.

Of these, the principal are only determined by careful computation, employing as data the quantities graphically measured on the indicator-diagram; others are at once seen by

the practised eye, demanding only an inspection of the figures shown on the card. An engine well adapted to its purpose, a perfect engine in the engineer's sense, will usually exhibit an early induction; wide port-opening; an admission-line closely approaching boiler-pressure, and nearly or quite horizontal; a sharp cut-off; an expansion-line closely approaching the common, or equilateral, hyperbola in form; a somewhat early and a prompt release or exhaust; a low and uniform back-pressure; and a compression carried up well toward initial pressure. These effects are obtained by giving some steam and exhaust lead—greater as speeds and pressures are higher—having good

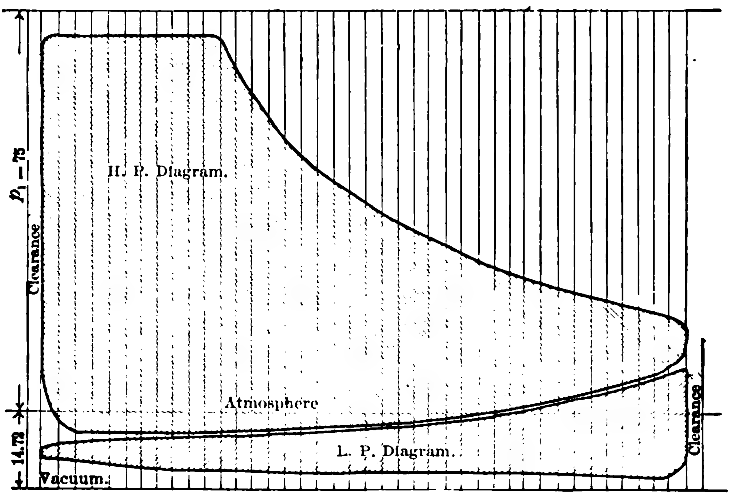

FIG. 207.—ACTION OF STEAM; WOLFF ENGINE.

area of ports, securing quick action of the expansion-valve, and a well-adjusted closure of the exhaust-valve. Any departure from these conditions is ordinarily to be taken as evidence of defective construction or adjustment.

200. Compound-engine Diagrams are precisely similar to that of a simple engine. Large steam-ports and a good expansion-gear bring the steam-line close up to that of boiler-pressure; a well-jacketed cylinder allows the expansion-line to

follow closely that laid down for the ideal engine; short and free ports between the two cylinders give an exhaust from the high-pressure and a supply to the low-pressure cylinder which are nearly coincident; and the two cards would, if reduced to a single diagram, exhibit a very close approximation to that which would have been constructed as the ideal diagram of this class of engine. Such satisfactory results are rare; and in most cases the differences between the actual and the ideal case are very marked, and are serious in their effect upon efficiency.

201. **The Construction of Diagrams** exhibiting the method of action of steam in the cylinders of the compound engine, as a preliminary to the settlement of the details of the design, is usually as below; these ideal or theoretical indicator-cards indicating the ideal action of the type of engine proposed, as modified by such conditions of operation as the designer can with more or less exactness define and represent. The separate diagrams appertaining to the two cylinders, high- and low-pressure, of an ideal compound engine, are to be combined in a manner indicated as follows,—to obtain a single diagram representing the complete cycle of changes of pressure and volume of the steam from the moment of entrance up to that of its discharge,—the Wolff type of engine being chosen for illustration: The steam being admitted into the smaller cylinder until it fills a volume, represented by $\overline{BC}$ in Fig. 208, the absolute pressure is represented by BO above zero on POQ.

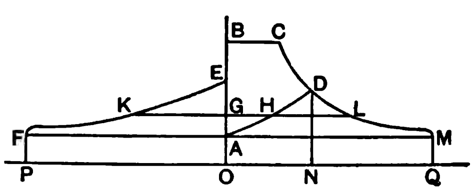

FIG. 208.—COMBINED DIAGRAMS.

The steam is then cut off, and it expands with a pressure gradually diminishing, as shown by the curve CD. DN being perpendicular to OQ, $\overline{ON}$ represents the space swept through by the piston of the small cylinder. Next a communication is opened between the small and the large cylinder; and the forward stroke of the large piston takes place at the same time with the return stroke of the small

cylinder. Thus the steam is driven before the piston of the small cylinder, and drives that of the larger cylinder, and it exerts more energy on the latter than it receives from the former, as the piston of the large cylinder sweeps through greater space; the difference between those quantities of energy is added to the energy exerted on the piston of the small cylinder. This action is represented by DA and EF; the ordinates of DA representing the backward pressures in the small cylinder, and EF the forward pressures in the large cylinder.

During return-stroke of the larger cylinder the steam is expelled, exerting a back-pressure, FA; while steam is admitted again into the small cylinder, and expanded during a new stroke of that cylinder.

Thus are obtained the diagrams $BCDAB$ for the small cylinder and $EFAE$ for the large, and the sum of their areas represents the energy exerted by the quantity of steam expended.

When the diagrams are to be used for the purpose of studying the relations between heat expended and work performed, it is best to combine them into one diagram, thus:

Draw a line KGH parallel to POQ, intersecting both diagrams, and lay off upon it $\overline{HL} = \overline{KG}$; and $\overline{GL} = \overline{GH} + \overline{KG}$ represents the total volume in both of the steam-cylinders, when its pressure is $\overline{OG}$; while L is a point which would have been reached had the action taken place in the large cylinder alone.

By drawing a number of lines, as KL, any number of points may be found to complete the *combined diagram* $BCDLMAB$, whose length $\overline{OQ} = \overline{OP}$ represents the volume of the large cylinder; and this diagram may be discussed as if it represented the action of the steam in the large cylinder only.

202. Special Applications of the indicator are of peculiar interest to the engineer. Valve-adjustment is often performed, and should always be checked, by the aid of this instrument. The application of the indicator to the steam-chest, and the comparison of its readings with those of the ordinary diagrams and of the steam-gauge at the boiler, will often reveal defects in the steam-passages or valve-action otherwise difficult of de-

tection; its use on the air-pumps of condensing engines and on the main pumps of pumping-engines similarly reveal anything objectionable in their construction and operation; and the motion being derived from the eccentric or the valve-mechanism when attached to the engine in the usual manner will permit a more minute examination of those phases of operation which are not easily studied on the common form of card. In some cases a continuous motion, derived from the crank-shaft, is adopted for this purpose.

In valve-adjustment, an inspection of the diagram shows the operation of the valve-mechanism as set. The necessity of adding lead or the reverse, of resetting the valves and eccentrics, is seen, and they are readjusted and diagrams again taken, these operations being repeated until the form of diagram desired is approximated as closely as is prrcticable.

203. **The Apparatus and Methods of Engine-trials** are necessarily somewhat different with differences of purpose and of data desired. They include such as aid in the direct measurement of the diagrams, and also instruments employed to measure the .speed of the engine and its fluctuations. Among the former is the planimeter; among the latter, speed-indicators, counters, and chronographs of various kinds. The methods of their use and of computations based on their work should always be such as will yield results of the greatest practicable exactness. The following are the principles and practice to be adopted wherever practicable, especially in trials having a scientific character and purposes of research.*

Before commencing, all apparatus to be used should be adjusted and carefully compared with standards, under the same conditions as in actual practice. The errors or constants of all instruments should be noted in the report of the test, and corresponding corrections made in the data obtained.

The instruments to be calibrated are:

(1) *Steam-gauge.*—Compare with mercury-column or with

* Compiled and condensed by Professor Carpenter, mainly from the Author's "Engine and Boiler Trials."

standard gauge, for each five pounds of pressure, reading both up and down throughout the range of pressures likely to be used in the test.

(2) *Steam-engine Indicator-springs.*—Put the indicator under actual steam-pressure and compare the length of ordinate of the card with the reading of the mercury-column for the same pressure. Take ten readings, both up and down, through an extreme range equal to one and a half times the number on the spring. The steam-pressure may be varied by throttling the supply and exhaust. The ordinate may also be compared by special methods with readings of a standard scale, the indicator being heated by the flow of steam through a rubber tube wound around it. (See "Engine Trials," p. 128.)

(3) *Speed-indicators.*—The accuracy can be determined by hand counting. (Ibid., p. 223.) For the best work chronographs should be used. (Ibid., pp. 229–240.) Continuous counters are necessary for accuracy in a long run.

(4) *Indicator Reducing-motion.*—This may be tested by dividing the stroke of the engine on the guides into twelve equal parts, and noting whether the card is similarly divided. It should be tested for both return and forward stroke. When the form of the card is considered, this is an important matter, as many reducing-motions distort its shape. (Ibid., pp. 160–180.)

(5) *Indicator-cords and Connections.*—See that the connecting cords do not stretch at high speeds, and that the drum-spring of the indicator has a proper tension and gives a correct motion of the drum.

(6) *Scales.*—Readings may be compared with standard weights.

(7) *Meters.*—To be calibrated by actually weighing the discharge under conditions of use. In case meters are used, temperatures of the water must be taken in order to obtain the weight.

(8) *Thermometers.*—Test the thermometer for freezing-point, by comparison with water containing ice or snow; test for boiling-point by comparison with steam at atmospheric pressure,

in special apparatus. The correct-boiling point to be determined by readings of the standard barometer. The other tests of the thermometer can in general be left to the makers of the instrument. In cases where great accuracy is required, the readings should be compared throughout the whole scale with the standard air-thermometer.

(9) *Pyrometer.*—Should be compared with a standard thermometer for the lower ranges of temperature, and with known melting-points of metals for higher. The correction may also be determined by cooling heated masses of metals in large bodies of water, and calculating the temperature from the known relations of specific heats.

(10) *The Planimeter* should be calibrated by making a comparison of its readings with a standard area. (Ibid., p. 219.)

PREPARATIONS FOR TESTING.

(1) *Weight of Steam.*—Prepare to weigh all the steam supplied the engine. This may be done by weighing all the feed-water supplied the boiler, provided there is no waste, nor other use of steam; or it may be done by condensing and weighing all the exhaust from the engine. In this last case especial precaution must be taken to prevent leaks, and the temperature of the condensed steam should be reduced to 110° F. before weighing. The weights may in some cases be determined from a meter reading.

(2) *Quality of Steam.*—Attach a calorimeter, which may be of the throttling or throttling-separator kind, to the main steam-pipe, just before it enters the engine. This attachment may be made to a one-fourth-inch pipe, cut with a long thread and extending three fourths across the main steam-pipe. This pipe should be provided with holes so that steam will be drawn from all parts of the main steam-pipe. For scientific purposes, make calorimeter determinations of the exhaust-steam, steam in steam-chest, average in cylinder and at compression. A special attachment for this purpose has been applied in some cases. (Ibid., p. 102.)

(3) *Leaks.*—The engine should be tested for piston-leaks by turning on steam with the piston blocked and cylinder-cocks opened on the end opposite that which steam is supplied.

(4) *Indicator Attachments.*—Arrange a correct reducing-motion. The lazy-tongs or pantograph is reliable for speeds less than 125 revolutions per minute, and can be easily applied. The pendulum pivoted above and furnished with an arc, although not perfectly accurate, is much used. (Ibid., p. 159.)

(5) *Absorption-dynamometer.*—Arrange a Prony brake to absorb the power of the engine, and make effective provision for keeping it cool. In many cases of commercial tests the power is absorbed by machinery or in useful work, and the efficiency is wholly determined by measurements of the amount and quality of steam and from the indicator-diagram. (Ibid., pp. 262–290.)

(6) *Weight of Coal.*—This is generally taken during an engine-test, but will be treated as pertaining to boiler-testing.

(7) *Clearance.*—The volume of clearance-space should be ascertained by placing the piston at the end of its stroke and filling the clearance-space with shot or water. Clearance is usually expressed in percentage of the volume swept through by the piston.

Remarks.—It will be found advisable to make a preliminary run of a couple of hours, before beginning the regular trial to ascertain if all the arrangements are perfect.

204. **The Measurements** taken in working up indicator-diagrams demand great care and accuracy. The figure to be measured is small; its bounding-lines often obscure and generally irregular; and the determination of its exact area, which is the usual problem, requires nice manipulations. An indicator-diagram represents the pressures, volumes, and work of the steam, or other fluid, at every instant throughout a single revolution of the engine, on *one* side of the piston. A pair of cards exhibits these quantities on both sides during one revolution. A series of such pairs exhibits the varying pressures and work of the engine at the several single revolutions to which they severally appertain. The average of the pressures

shown on one card is the mean pressure for a single revolution on one side the piston; the average obtained from a series of diagrams gives a mean of the pressures, for the period covered, with a degree of approximation dependent upon the number of diagrams and the uniformity of action of the engine. By taking diagrams with sufficient frequency, any desired accuracy may be attained. In practice, they are often taken as seldom as once an hour, and, on trials of importance, sometimes as often as every fifteen minutes. At sea, it is customary on naval vessels to take a set of diagrams once a day.

Since the diagram only gives the pressures, the other factors of work and of power must be determined otherwise. The indicated work of the engine at each stroke is the product of the net intensity of pressure on its piston by the volume traversed. The power is the work done by the engine in the unit of time; in British measures,

$$\text{H. P.} = 2\frac{plan}{33000};$$

where p is the average net effective pressure of the steam as shown by the indicator, l is the length of stroke, a is the effective area of piston, n is the number of revolutions per minute. Pressures are here, as usual, measured in pounds on the square inch, areas of piston in square inches, the stroke in feet, and work in foot-pounds per minute. Of these quantities, all but p are obtained by direct measurement and by observation. The pressure p is the one quantity obtained by the use of the indicator. The method of determination is to measure the area of the diagram, divide by its total length, and thus obtain in the quotient its mean altitude. This being multiplied by the scale of the spring and of the ordinates gives the mean pressure.

The following, from Rankine, illustrates a good method of record and computation.* Ten ordinates are measured, and the results for both cylinders of a compound engine are given.

* Steam-engine, p. 51.

COMPOUND-ENGINE DIAGRAMS.

Ordinates.	First Cylinder.		Second Cylinder.	
	Top.	Bottom.	Top.	Bottom.
b_0	27	36	16.0	12.4
b_{10}	13	12	2.0	3.8
Sum	40	48	18.0	16.2
Half sum	20	24	9.0	8.1
b_1	83	97	10.5	10.8
b_2	91	96	8.5	9.0
b_3	91	84	7.5	8.0
b_4	64	64	7.0	7.1
b_5	57	57	6.6	6.7
b_6	53	46	6.2	6.0
b_7	42	40	6.0	5.6
b_8	35	32	5.1	5.4
b_9	22	22	4.5	5.0
Sum	558	562	70.9	71.7
Sum ÷ 10 = M. E. P	35.8	56.2	7.09	7.17
Mean, top and bottom	56.0		7.13	
× area of piston, sq. in	345		1380	
Mean effort, in lbs	19320		9839.4	
× stroke, in feet, 2½ × revolutions per minute, 52½ × 2 =	262.5		262.5	
Ft.-lbs. per minute	5071500		2582842.5	
Total	7,654,342.5			
÷ 33,000 =	232 I. H. P.			

Those mean pressures are found by a process which may be algebraically represented as follows:

Divide the length of the diagram into any convenient number, n, of equal parts, and measure the ordinates at the two ends and the $n - 1$ points of division; so that ordinates are measured from $n + 1$ equidistant points.

Let p_0 be the first, p_n the last, and p_1, p_2, etc., the intermediate ordinates of the upper curve; let p_0' be the first, p_n'

the last, and p_1', p_2', etc., the intermediate ordinates of the lower curve; let p_m denote the mean forward pressure, p_m' the mean back pressure, and $p_m - p_m'$ the mean effective pressure. Then

$$p_m = \frac{1}{n}\left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.}\right);$$

$$p_m' = \frac{1}{n}\left(\frac{p_0' + p_n'}{2} + p_1' + p_2' + \text{etc.}\right);$$

$$p_m - p_m' = \frac{1}{n}\left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.} \dots - \frac{p_0' + p_n'}{2} - p_1 - p_2 - \text{etc.}\right).$$

The mean pressures thus obtained are then employed as factors in the expression for the power of the engine, and their comparison with that computed for the ideal case will give some idea of the character of the expansion-line.

The mean effective pressure may be computed at once by measuring a series of equidistant breadths of the diagram; the mean of which breadths represents the mean effective pressure. That is, let b_0 be the first, b_n the last, and b_1, b_2, etc., the intermediate breadths.

Then

$$p_m - p_m' = \frac{1}{n}\left(\frac{b_0 + b_n}{2} + b_1 + b_2 + \text{etc.}\right).$$

The effective energy exerted by the steam on the piston during each revolution is the product of the mean effective pressure, the area of the piston, and the length of stroke, or

$$(p_m - p_m')As;$$

and if N be the number of double strokes in a minute, the power in foot-pounds per minute is

$$p_m - p_m')ANs;$$

from which the indicated horse-power is found by dividing by 33,000.

The presence of the piston-rod or of a "half-trunk" on one side the piston produces a difference of areas which, in the latter case, is of considerable magnitude. Where measured separately, if A_1 and A_2 are the two areas, the power is found to be

$$I.H.P. = \frac{(p_1 A_1 + p_2 A_2)ln}{33,000},$$

l and n being the length of strokes of piston and the number per minute.

The following diagram will be found to give approximately the total mean effective pressure of steam in an engine-cylinder, when the initial steam-pressure and the point of cut-off are known.

The left-hand figures are the initial steam-pressures (above the atmosphere), and the upper horizontal column, the points of cut-off. Directly under this column is the mean effective pressure. To determine, for example, the mean effective pressure obtained with 90 pounds of steam, cut-off at one quarter, find 90 in the initial-pressure column, and follow the line to the right until it intersects the oblique line that corresponds to $\frac{1}{4}$ cut-off. Read the mean effective pressure from the figure directly above, which in this case is 49 pounds. On the lower scale the figure that corresponds with this point of intersection, the percentage of gain in power that a vacuum will give in an engine using 90 pounds of steam, cut-off at one quarter,

will be seen. Thus in this instance the value is found to be between 25 and 30 per cent of the power of the engine running non-condensing.

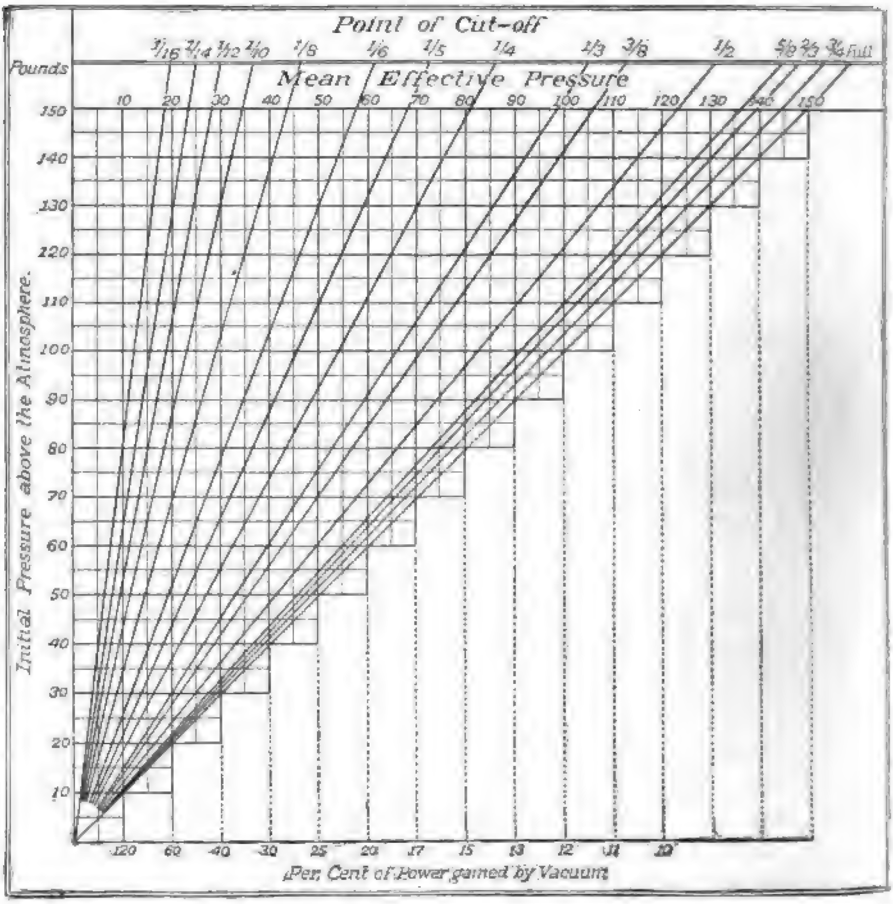

FIG. 209.—MEAN EFFECTIVE PRESSURES FOR NON-CONDENSING ENGINES.

205. Tachometers, Speed-indicators, Chronographs, and Counters are instruments of various kinds and classes used by the engineer in determining the speed of the engine, the second of the essential factors obtained by observation in measur-

ing its power. When indicator-diagrams are taken, the speed of rotation of the engine is taken as nearly simultaneously as possible. This measurement is commonly taken with one of the small hand speed-indicators, the spindle of which is applied to the centre in the end of the main shaft and there held a quarter or half minute, a full minute, or more, as less or greater accuracy is desired and as the speed of the engine is greater or less.

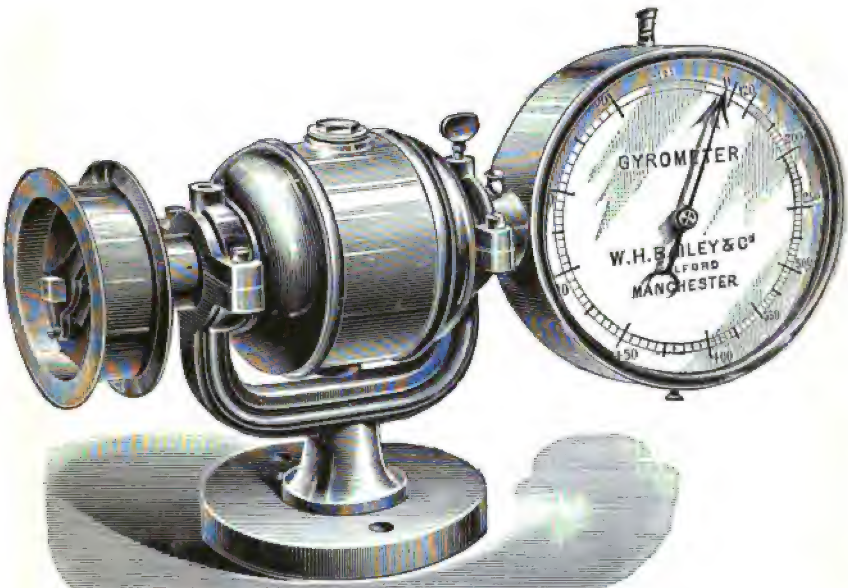

FIG. 210.—SPEED-INDICATOR.

The chronograph was first applied to measure the variation of velocity of the steam-engine by a committee of the British Association in 1843–4, in their determinations of the speeds of piston of the Cornish engine. It has been applied by Mr. Woodbury, as early as 1873, to pumping-engines, determining the flucuations of velocity of fly-wheel, and, later, by Mr. Eckart in observing similar fluctuations of pump-rod speed, at great depths in the Comstock lode in Nevada. The latter

found it practicable to use the instrument at speeds of from 80 feet per minute up to 1400 feet. Messrs. Dix and Mack, under the supervision of the Author, applied the same instrument, still later, to the "high-speed" engine, making 250 and more revolutions per minute.

The frontispiece to this chapter, Fig. 188, exhibits the method of attachment of the instrument to a "high-speed," direct-acting, horizontal engine of common type. *A* is the steam-cylinder, *B* the engine-frame, *C* the end of the main shaft, *D* the balance-wheel *E*, the brake-pulley, with strap *F*, and scale weighing the turning effort at *G*. On the extremity of the shaft, a coupling *H* is attached which drives the chronograph *I* through a slender rod seen connecting them. The revolving cylinder, on which the paper is smoothly stretched to receive the record, is seen at *J*, and the stylus or pen is at *K*. The whole is mounted on a firm support as *L*.

When in operation, the cylinder is turned by the engine, instead of its usual motor apparatus ,and the pen, slowly traversing the cylinder, produces a closely compressed helix. At regular intervals a circuit is made and broken by the standard clock or other timing instrument, and the line is given a V-shaped jog which marks the time-interval on the cylinder. The adjustment should be such that, at normal speed, these breaks should occur at precisely the same points in the circumference of the chronograph cylinder at each of its revolutions or at each tenth or other fraction of a revolution, as may be determined upon. Any acceleration or retardation will then be exhibited by the production of the break in advance of, or behind, its normal position. In the first case all such breaks fall into straight lines along the cylinder, parallel to the axis; in the latter case they will fall into regular helical, or curvilinear, or irregular lines, accordingly as the acceleration or retardation is uniform, smoothly variable, or irregular. The inclination of the lines, or of the tangents to the curves so produced, to the axis is thus a measure of the change of speed.

The engraving exhibits the chronograph as used by Mr. Eckart. The reference-letters are as follow:

CC—Cast-iron base-plate, covered with sheet-brass, upon which the mechanism is secured.

B—Metal frame containing gearing for driving drum *A* and escapement-wheel *b*; motion communicated by means of adjustable weights *D*.

AA—Light brass drum, accurately balanced, revolving on friction-rollers 8, 8, at both ends.

ff—Parallel guide-bars upon which the tracing-point h_0 and its carriage travel back and forth, receiving motion in one direction from the engine or other moving parts through the cord *P*, passing through the bars *f*, and attached to the tracing-carriage; the return motion is derived from a coiled spring in the spring-drum *C*.

ee—Small electro-magnets on tracing-carriage for raising the tracing-point h_0 off the paper and replacing it at any desired point to be especially observed.

d—Electro-magnets on separate carriage *kk*, adjustable on parallel bars *f*, operating the steel tracing-point *g*, at-

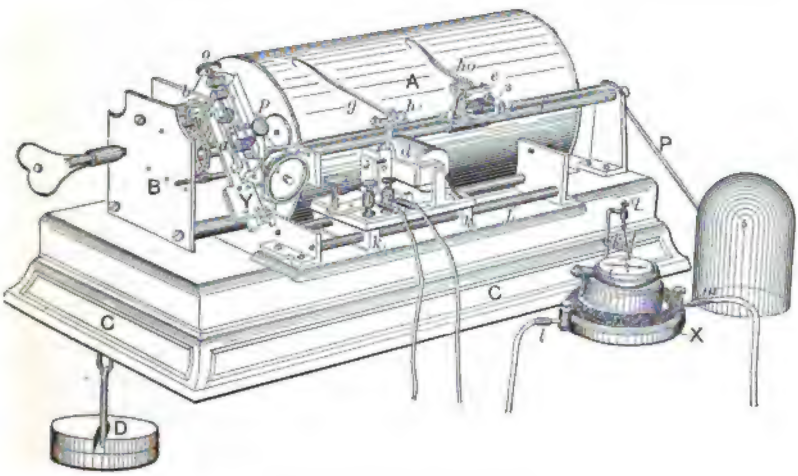

FIG. 211.—THE CHRONOGRAPH.

tached to the armature of *d*, for the purpose of recording seconds on the margin of the paper or at other parts of same, as required.

i—Chronoscope or watch supported on frame *x*, the second-hand of which swings the light platinum wire *J*, breaking contact with the insulated wire *k*, thereby breaking circuit with *d* and recording seconds through the tracing-point *g* on the paper.

q—Adjusting screw for the wire *J*.

a—Steel spring of escapement. This spring is securely clamped in *Y*, its flexibility being controlled to a certain extent by means of the thumb-screws *o* and *p*.

This instrument was found to give with great exactness the fluctuations of piston-speed in a pumping-engine at 80 feet per minute, and for a hoisting-engine at 1400 feet. It is thus peculiarly well adapted for pit-work.

Instead of, as is usual, employing a clock pendulum to mark time and give the velocity-curve of the engine, a portable time-keeper is here used. This is the common form of "timer," designed especially for timing in trials of speed of vessels, or on the race-course. The hand of the instrument, revolving once per second, breaks the circuit, and the stylus or pen *g* is caused to mark the interval. The stylus or tracing-point barely touches the lamp-black, being counterbalanced in such manner as to remove the coating without bearing perceptibly upon the paper, producing a fine white line on the black surface. In fitting the paper, it is cut slightly longer than the circumference which it is to cover, and lapping the edges and gluing them together, the lap is carefully sandpapered to as nearly as possible uniform thickness at the joint and elsewhere. The surface is then smoked, and is ready for use.

Speed-indicators are sometimes made to give a permanent record, in a form suitable for preservation, of the fluctuations of speed of engine. The Mosscrop Speed-recorder, for example, consists of a clock which causes a sheet of paper to traverse at uniform speed, while the engine drives a small revolving pendulum connected in such manner as to cause a pencil to indicate continuously the speed of revolution.

The diagram illustrates the record of a Mosscrop Speed-recorder for several consecutive hours. The distance between

vertical lines represents five minutes, and between horizontal lines three per cent variation of speed.

The diagram was made from a 9×12 Ball engine at the Boston Fair of 1883. The engine was driving four dynamos, which were thrown on or off at pleasure. The record began at 5.10 P.M., and is shown until nearly 9 o'clock. Examination of the diagram shows that at no time did the speed vary more than a fraction of one per cent.

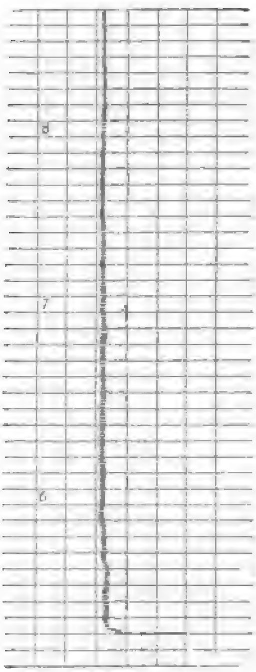
5 P.M. 9. 3. '83.

FIG. 212.—RECORD OF SPEED.

The Hand Speed-indicator is made in various forms. One which the Author has found very satisfactory answers equally well in whichever direction the engine may turn, is convenient in use, and gives reliable results. The Author has often flattened the point and given it a semicircular sharp-edged termination, to obtain a more secure hold on the centre of the shaft.

The usual method of counting the revolutions of the engine, by means of the hand speed-indicator or the registering "counter" attached to the machine, gives the mean speed for a certain time—as for a minute—for which the count is taken. The use of the chronograph in the manner just indicated gives a measure of the *rate* or speed for the instant, for each revolution. To ascertain the rate during successive *portions* of the revolution, the method of Woodbury or of Eckart may be adopted.

206. The Steam or Water Consumption of the engine cannot be exactly ascertained by the use of the indicator. The indicator does exhibit, however, the pressure and volume of steam actually present as steam, at each instant in the steam-cylinder, and it thus becomes easy to compute its weight and to obtain a measure of the quantity thus shown by the indi-

cator for comparison with the total quantity supplied by the boiler, and thus to ascertain the losses by condensation and leakage. The pressure being shown on the diagram at every point in the stroke, the "steam-tables" give the corresponding specific weights, the weights per unit of volume; and the space traversed by the piston up to the given point, plus the clearance-space, measures the actual volume; the latter quantity, multiplied by the specific weight, is the weight of uncondensed steam present in the cylinder at the specified point. The mean weight per stroke, multiplied by the number of strokes, being compared with the total weight supplied by the boiler in the same time, as shown by the log of the boiler-trial, the difference is the waste by internal condensation and leakage.

When the problem to be solved is, as usual, the determination of the efficiency of any actual engine, as distinguished from the simple thermodynamic efficiency of the ideal engine, the indicator aids the engineer in its solution by showing the precise quantity of steam present at every instant during the stroke, and hence the quantity of water present at the same time; the sum of these two weights being always, if the piston and valves are tight, equal to the weight of feed-water passing through the boiler and entering the engine as a mixed working fluid.

In the appendix to Part I will be found Mr. Thompson's table of factors for computations of water-consumption in the steam-engine, so far as shown by the indicator.*

In their use a vertical line is drawn at each extremity of the diagram, and the expansion-line is continued to intersect that at its end. The "absolute" pressure is measured at this intersection, which pressure is that which the steam would have had if expanded to fill the cylinder instead of being earlier exhausted. Finding the number in the table corresponding to this absolute pressure, it is to be divided by the mean effective pressure obtained from the diagram; and the

* See Hemenway's Indicator Practice; N. Y., J. Wiley & Sons, 1886; and the Author's Engine and Boiler Trials.

quotient is the steam-weight accounted for by the indicator, uncorrected for compression. Finding the ratio of volume of cushion-steam at the same pressure on the diagram to the cylinder-volume, the product of their ratio into the weight just obtained gives a smaller, and the correct, weight of uncondensed steam at the end of the stroke.

The following are the data obtained by a duty-trial reported by Remington & Henthorn, on an engine of the "cross-compound" type, elsewhere illustrated, as in operation at Bristol, R. I., in the Namquit Mills. These figures may be taken as very near the practical minimum for contemporary engines of this class, and under the stated conditions of operation:

Quantity of water discharged from the condenser during the ten hours........lbs.	27,556	
Quantity of water discharged from the high- and low-pressure cylinders and receiver during the ten hours........lbs.	3,439	
Total quantity of water required to develop power of engine for the ten hours...lbs.		30,995
Quantity of water required per hour..lbs.		3,099.5
Average indicated horse-power, determined from thirty (30) sets of cards:		
Front end of 18-inch cylinder, horse-power	56.5	
Back end of 18-inch cylinder, horse-power	53.15	
Horse-power of 18-inch cylinder.........		109.6
Front end of 32-inch cylinder, horse-power	60.2	
Back end of 32-inch cylinder, horse-power	63.8	
Horse-power of 32-inch cylinder..........		124.1
Total average indicated horse-power.....		233.7
Total water per hour per indicated horse-power:		
$\dfrac{3{,}099.5 \text{ lbs. per hour.}}{233.7 \text{ indicated horse-power}}$.		13.26
Average steam-pressure, corrected.......		98.

Cylinder-condensation and Leakage produce variations in the diagram, as obtained, which differently affect the different parts of the curve. Leakage can usually be eliminated, and always should be before the engine is set at work regularly. The first-named waste is usually irremediable. When the exact measure of the quantity of steam expended is obtained by a boiler-trial, it is easy to trace these variations, as in the diagram here given, as taken from the engine and worked up by the late Professor C. A. Smith, in which illustration the diagram which should have been produced by the same steam, had there been no initial condensation, is shown with the real diagram.*

This indicator-diagram is an unusually good sample, as to form, and was taken from the St. Louis high-service pumping-engine, a machine of 705 I. H. P., 85 inches diameter of cylinder, and 10 feet stroke of piston, making $11\frac{1}{2}$ revolutions per minute. Taking measures of the abscissas of the two dia-

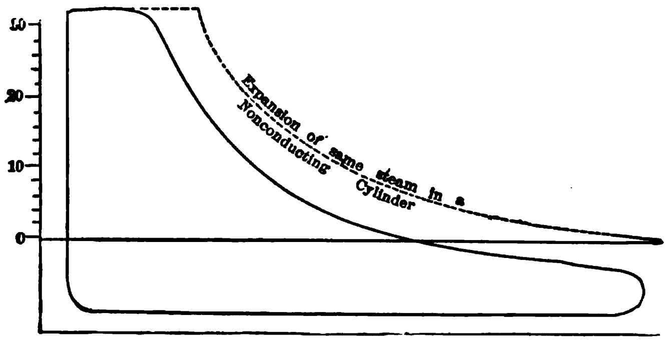

FIG. 213.—THE REAL AND THE IDEAL CARD.

grams, it is seen that the condensation amounts to from about 30 per cent as a minimum to 50 per cent as a maximum, so far as measurable, the actual card illustrating the expansion in a metallic cylinder of the steam, which would have given the larger diagram in an ideal engine with its non-conducting cylinder. When the two lines continue so far separated, it is

* Steam-making; p. 91.

an indication of large initial condensation and correspondingly great re-evaporation after the exhaust-valve opens; as the initial condensation is due to, and is proportional to, this re-evaporation. In most cases, however, the engineer, unable to determine these data, assumes the point of release, or the point of intersection of the expansion-line prolonged with the ordinate at the extreme end of the diagram, as that of coincidence of the ideal and the real curve, and draws the hyperbolic curve backward from that as a given point, in the manner already described.

A complete investigation of these wastes may be made by a careful experimental application of Hirn's analysis.

The extent to which this waste affects small engines, particularly, has already been fully shown in Part I, and is further illustrated by the following figures, obtained in regular work from a small engine driving the machinery of an electric-lighting station:

WATER-CONSUMPTION AT LOW POWERS.

Hour.	D. H. P.	I. H. P.	Water per H. P.	Water per available H. P.
6– 7 P.M.	21.5	26.2	25	28
7– 8	43.1	47.8	27	30
8– 9	43.1	47.8	27	30
9–10	21.5	26.2	25	28
10–11	9.	13.7	29½	43
11–12 M.	4.6	9.3	30½	55
12– 6 A.M.	2.8	7.5	31½	80

In this case the steam-pressure was 90 lbs., and the engine of a size suited for the heavier loads.

407. "Hirn's Analysis," already described,* is illustrated, in its application to scientific investigation, by the following, as reported at the close of this research, and as subsequently published: †

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* Part I, Chap. V, § 131.

† Proc. U. S. Naval Institute, vol. XVII, No. 3, 1891.

It had been found that it was possible to reduce the heat-transmitting and storing power of surfaces of cast-iron in presence of steam of higher temperature by either simple oxidation and conversion into a sponge of metallic iron filled in with the graphite of the casting, or by covering it with some non-conducting varnish, or by a combination of both methods, as proposed originally by the Author.* Either process was found to produce a gain of about 40 per cent, and the two about 60 per cent, under the best observed conditions.

It was expected that this treatment, applied to the inner surfaces of a steam-cylinder, would, to the extent to which the iron was so treated, diminish by at least 40 per cent the internal wastes of the engine. Here the varnish is relied upon to retard interchange of heat between the steam and the metal, and the alteration of surface produced by the acid is expected to render the coating of varnish permanent.

When investigating the causes of boiler-explosions, it was found that simply oiling the surface of the plate reduced heat-transmitting power 10 per cent.

Experiments were made upon such an engine in the following manner:

The engine was first carefully overhauled; its valves and piston were seen to be perfectly tight, and the engine otherwise in good order. A very careful trial of the machine was then made, and the results of the test—to be presently given—showed that it was in a condition of high efficiency. The surfaces of the cylinder and heads, so far as practicable, i.e., where not exposed to the rubbing action of piston and rings, were next treated in the manner shown by preliminary investigation to be most effective in securing the desired alteration of the surface. It was then given a coat of drying-oil, and allowed one day for its oxidation and the formation of a varnish.

A second trial was made and the difference in efficiency

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* Trans. A. S. C. E., vol. XXIII, No. 144, 1890; Trans. A. S. M. E., vol. VII, p. 575; Ibid., vol. XII, p. 174, *et seq.*

noted—a difference corresponding very closely with that previously anticipated, and computed as probable.

The engine employed was built at the request of the Author as an "experimental engine." Its steam-cylinder is 6 inches in diameter and the stroke of piston 8 inches. Following are the logs obtained as registered on the usual forms. The first is that obtained before, the second that given after, the engine had been treated as described. The steam was measured by employing a surface-condenser. This condenser was that in regular use, and was tight and efficient.

A study of these logs indicates that the engine, probably in consequence of differences in effectiveness of lubrication, ran a trifle faster on the second day than on the first; the machine having, meantime, been taken apart and reassembled. The differences between indicated and dynamometric power amounted to 0.9 H. P. the first day and 0.6 the second; the total being highest for the indicated power the first day, and for brake-power the second. The engine, after being overhauled, was less subject to friction-losses. The condenser was carried a little colder the second day, which would tend to exaggerate cylinder-condensation. On the whole, the conditions were practically the same, so far as they affect the matter in hand; and the comparison is a fair one.

The indicator-diagrams taken were substantially alike on both days.

The power was well distributed between the two ends of the cylinder; the mean pressures differing but two pounds.

The investigation of Dwelshauvers-Dery consists in the measurement, by calorimeter, indicator, and brake, of the quantity of heat-energy brought to the engine by the steam; the quality of the fluid at entrance into the steam-chest; the same quantities and the dynamic energy developed, and its distribution, as the engine passes through a complete cycle; the mean, however, being usually taken for a considerable number of such cycles or revolutions. Thus it becomes possible to determine just what is the method of variation of the proportions of steam and water in the mixture entering the engine,

LOG OF TEST OF ENGINE.

Number.	Time.	Revolutions. Continuous Counter.	Revolutions. Speed-indicator.	Gauge-readings. Pounds. Boiler.	Pounds. Steam-pipe.	Pounds. Steam-chest.	Inches Hg. Exhaust.	Inches Hg. Condenser.	Inches Hg. Barometer.	Temperatures. External Air.	Engine-room.	Condensed Steam.	Exhaust-steam.	Injection-water.	Discharge-water.
1	2.45		200			72.5	22	23^3			77.8	110.4	158.	43.6	105.8
2	3.00		201			72.0	22	23^3		66.2	79.4	111.6	158.5	43.7	108.5
3	15		199			70.0	22	22^3			80	123.6	161.	44.0	110.6
4	30		198			72.0	22^1	24^2			79.4	113.6	157.	42.8	83.3
5	45		197			70.0	23	24^1			80.6	99.0	157.	42.9	83.4
6	4.00		203			71.0	22^2	24^2		66.4	79.6	110.8	157.	43.4	98.6
7	15		202			72.0	23	25^1	29.276		81.5	104.2	155.5	43.0	98.2
8	30		196			72.0	23	24			81.4	104.0	158.	42.9	91.4
9	45		202			68.0	23	25^2			82.4	97.6	158.	42.9	85.6
10	5.00		200			62.0	23^2	25^1			83.0	92.2	158.5	43.0	83.1
11	15		180			60.0	22^2	23^2		67.0	82.	89.0	157.	43.0	85.8
12	30		200			65.0	24	26			81.2	97.8	151.5	43.0	83.2
Mean			198.17			68.88	22.73	24.42		66.5	80.7	104.5	157¼	43.19	93.14
Corrected Mean.			198.17			64.80 79.155 abs.			14.355 lbs.	65°.865	79°.923	103°.475	157°.250	42°.758	92°.219

Number.	Calorimeter. Steam-pipe.	Calorimeter. Steam-chest.	Calorimeter. Cylinder.	Calorimeter. Exhaust.	Weights. Condensed Steam.	Weights. Feed-water.	Weights. Injection-water.	Head. M. E. P.	Head. I. H. P.	Crank. M. E. P.	Crank. I. H. P.	I. H. P. Total.	Brake. Load.	Brake. D. H. P.	Brake. Scale of Spring.
1			263.5					29.0	3.4	29.2	3.3				
2			272.5					27.3	3.2	29.7	3.4				
3			277.0					29.0	3.4	29.3	3.3				
4			280.0					30.4	3.5	32.5	3.6				
5			280					29.3	3.4	31.5	3.5				
6			280					29.7	3.5	30.2	3.4				
7			281					31.2	3.7	29.5	3.4				
8			281					29.7	3.4	31.2	3.4				
9			277					27.0	3.2	27.6	3.1				
10			276					28.3	3.3	29.0	3.3				
11			273					26.5	2.8	26.5	2.7				
12			271					27.0	3.1	28.5	3.2				
Mean			276												
Corrected Mean.			276°.828		714.750		13873.41		3.3152		3.3054	6.6206	57	4.7100	40

LOG OF TEST OF ENGINE.

Number.	Time.	Revolutions. Continuous Counter.	Revolutions. Speed-indicator.	Gauge-readings. Pounds. Per cent Entrained water.	Gauge-readings. Pounds. Steam-chest.	Gauge-readings. Inches Hg. Exhaust.	Gauge-readings. Inches Hg. Condenser.	Gauge-readings. Barometer	Temperatures. External Air.	Temperatures. Engine-room.	Temperatures. Condensed Steam.	Temperatures. Exhaust-steam.	Temperatures. Injection-water	Temperatures. Discharge-water.
1	7.15		206	2.66	73	23^{1}	24^{2}		58	80	100	152	53	99
2	30		205	1.50	67	23^{2}	24^{3}			78	103	151	53	85
3	45		207	1.70	72	23^{3}	25			80	97	152	52.5	85.5
4	8.00		205	2.50	72	23^{3}	25			80	95	151	52.5	88.5
5	15		206	2.45	73	23^{2}	25	29.132	55	80	95	152	52.5	87.5
6	30		207	2.70	71	23^{3}	25			80	95	152	52.5	88
7	45		207	2.16	67	23^{3}	25¼			79	95	152	52.5	87
8	9.00		207	2.05	67	23^{3}	25¼		54	79	95	152	52.5	91
9	15		208	2.05	67	23^{3}	25¼			79	95	151	52.5	87
10	30		207	1.33	68	23^{3}	25⅜			81	94	150	52.5	87.5
11	45		207	1.00	70	23^{3}	25⅜		55	82	93	151	52.5	86.5
Mean...			206.56	2.01	69.73	23.66			54.5	79.8	96.0	151.15	52.76	88.41
Corrected Mean.			206.56	$X = 97.99$	69.40 83.700 abs.			14.300 lbs.	54°.50	79°.80	95°.06	151.66	52.065	87.555

Number.	Calorimeter. Steam-pipe.	Calorimeter. Steam-chest.	Calorimeter. Cylinder.	Calorimeter. Exhaust.	Weights. Condensed Steam.	Weights. Feed-water.	Weights. Injection-water.	Head. M. E. P.	Head. I. H. P.	Crank. M. E. P.	Crank. I. H. P.	I. H. P. Total.	Brake. Load.	Brake. D. H. P.	Brake. Scale of Spring.
1		228	261					26.5	3.2	29.7	3.5				
2		246	260					26.0	3.1	30.0	3.5				
3		245	261					26.4	3.2	27.8	3.3				
4		230	261					26.5	3.2	29.7	3.4				
								Indicator spring slipped.							
5		232	263												
6		226	261					24.6	3.0	27.0	3.1				
7		234	260					26.4	3.2	28.0	3.3				
8		236	260					26.0	3.1	27.5	3.2				
9		236	261					26.3	3.2	27.0	3.2				
10		250	261					27.1	3.3	29.7	3.5				
11		258	266					28.1	3.4	29.7	3.5				
Mean...															
Corrected Mean.			261.40		585 lbs.		15229.06 lbs.		3.17631		3.33507	6.51138	57.	4.90943	40

and step by step as it passes on into the condenser; to ascertain the quantities of heat converted into work, the amount wasted by external conduction and radiation, and the quantities lost and gained, from point to point in the engine-cycle, by the process of cylinder-condensation; and the comparison of the power shown by the indicator with that measured by the Prony brake gives the amount of dynamic energy lost by friction in the engine. The whole history of the energy supplied, as it streams through the engine and is distributed in its various forms of thermal and dynamic, and of useful and wasted power or energy, is thus given; and thus tracing its progress and its final disposition, the characteristic differences of engines, or of the same engine under differing conditions, may be ascertained. The data were recorded on blanks of the form here illustrated and the details and symbols are explained on each line. It is thus easy to ascertain precisely how the heat and the steam performed their work, and, at the same time, were usefully transformed or were wasted from their entrance into the engine to their discharge from the condenser.

The first set of figures recorded are the reduced observations from the logs.

DATA AND RESULTS.—FIRST TRIAL.

Engine, slide-valve, throttling. Diameter cylinder, 6".06. Length stroke, 8". Diameter piston-rod, 1"$\frac{3}{16}$.

Volume head end....	.13354
Volume of cylinder, crank end..............	.12921
Volume clearance, cu. ft., head...............	.01744
" " " crank..............	.01616
Pressure by gauge, steam-chest.............	64.80
Clearance in per cent of stroke...	13.06
" " " "	12.51
Barometer....	29."276
Pressure, absolute, steam-chest.............	79.155
Revolutions per hour.....................	11,890.20

Quality of steam in steam-pipe,	
Quality of steam in compression	1.0205
Weight of condensed steam per hour	259.92
Boiling temperature, atmosphere pressure	210.70
Steam used during run, pounds	716.4240
Quality of steam in steam-chest	.9941
Quality of steam in exhaust	.9021
Pounds of wet steam per stroke:	
Head	.0109707
Crank	.0109383
Temperatures condensed steam	$103^\circ.475 = Sg + 32$
Temperatures condensing water, cold,	$42^\circ.758 = Si + 32$
Temperatures condensing water, hot,	$92^\circ.219 = Sk + 32$
Pounds of condensing water, per hour	5044.878;
per stroke	$\begin{cases} H, .212016 \\ C, .212274 \end{cases}$
Cut-off, crank end per cent of stroke	20.544
" head end, per cent of stroke	18.963
Compression, crank end, per cent of stroke	52.341
" head end, per cent of stroke	39.770
I. H. P. $\begin{Bmatrix} H, 3.3152 \\ C, 3.3054 \end{Bmatrix}$	6.6206
Brake horse-power	4.7100
Release, crank end	93.958
Release, head end	94.971
Pounds of steam per I. H. P	39.351
Pounds of steam per brake H. P	55.314

DATA AND RESULTS.—SECOND TRIAL.

Pressure by gauge, steam-chest	69.40
" absolute, "	83.700
Revolutions per hour	12,393.60
Quality of steam in steam-pipe:	
Quality of steam in compression	1.020

Barometer	29''.132
Boiling temperature, atmosphere pressure	210°.62
Steam used during run, pounds	586.7041
Quality of steam in steam-chest	.9799
Quality of steam in exhaust	.86209
Weight of condensed steam per hour	234.000
Pounds of wet steam per stroke:	
Head	.0091623
Crank	.0097722
Temperatures condensed steam	95°.060 = Sg + 32
Temperatures condensing water, cold,	52°.065 = Si + 32
Temperatures condensing water, hot,	87°.555 = Sk + 32
Pounds of condensing water, per hour	6091.62;
per stroke	H, .237114; C, .254396
Cut-off, crank end, per cent of stroke	21.843
" head end, per cent of stroke	19.143
Compression, crank end, per cent of stroke	52.153
" head end, per cent of stroke	37.714
I. H. P. {H, 3.17631; C, 3.33507}	6.51138
Brake horse-power	4.90943
Release, crank end	94.977
Release, head end	95.714
Pounds of steam per I. H. P.	36.0413
Pounds of steam per brake H. P.	47.8022

The summary of data and the results, shown in the second of the following tables, constitute an interesting study, apart from their relation to the problem here solved. The table gives percentages of total heat, steam, and energy entering the engine from the boiler. On both occasions this steam was practically dry, the water constituting but one and a half to two per cent of the whole. At cut-off, this figure becomes from forty to nearly fifty per cent, condensed at that point.

DATA AND RESULTS PER 100 STROKES.

Quantities.	Symbols.	Formulæ.	1st Trial. Head.	1st Trial. Crank.	2d Trial. Head.	2d Trial. Crank.
Steam used by calorimeter............... lbs.	M_x		.00490		.00545	
Steam from boiler........ .... "	M		1.09707	1.09383	.91623	.97722
Steam in clearance.......... .... "	M_0	$100\dfrac{V_c+V_0}{C_0X_0}$	.16458	.14414	.14346	.14540
Steam, total............................ .. "	$M+M_0$		1.26165	1.23797	1.05969	1.12262
Heat of condensed steam............... ...	K'	$(M-M_x)S_g$	78.0629	78.1815	57.4338	61.6235
Condensing water..... ....lbs.	G		21.2016	21.2274	23.7114	25.4396
Heat given to condensing water......... ...	K	$G(S_k-S_i)$	1048.6523	1049.9284	844.5176	902.8514
Heat supplied to engine....	Q	$M(XL+S)$	1285.2743	1281.4785	1062.8957	1133.6487
Sensible heat at admission.	H_0	M_0S_0	37.9982	33.4579	32.5093	33.8271
Internal " " "	H_0'	$100\dfrac{V_c+V_0}{C_0}I_0$	143.5501	125.5798	125.5173	126.5534
Sensible heat at cut-off.....................	H_1	$(M+M_0)S_1$	338.3488	328.2161	281.9831	295.4915
Internal " " "	H_1'	$100\dfrac{V_c+V_1}{C_1}I_1$	545.5026	528.1695	534.8858	531.3785
Sensible heat at release........	H_2	$(M+M_0)S_2$	254.4084	254.0300	206.1891	227.8784
Internal " " "	H_2'	$100\dfrac{V_c+V_2}{C_2}I_2$	701.1183	706.5868	626.7889	688.2081
Sensible heat, beginning of compression....	H_3	$(M_0+M_x)S_3$	21.3924	19.0826	19.5355	20.8868
Internal " " " "	H_3'	$100\dfrac{V_c+V_3}{C_3}I_3$	113.9140	103.6313	131.0014	131.9348
Cylinder loss during admission..............	Q_a	$Q+H_0+H_0'-H_1-H_1'-\dfrac{W_a}{778}$	551.1356	549.8245	370.6544	430.0416
" " " expansion.............	Q_b	$H_1+H_1'-H_2-H_2'\dfrac{W_b}{778}$	− 135.8615	− 163.1786	− 75.1285	− 147.0454
" " " exhaust.......	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-\dfrac{W_c}{778}$	− 303.3649	− 285.3828	− 214.9133	− 193.6765
" " " compression...	Q_d	$H_3+H_3'-H_0-H_0'-H_x-H_x'-\dfrac{W_d}{778}$	− 29.7279	− 18.6036	9.0961	11.4304
Heat discharged, and work	B	$K+K'+H_x+H_x'+\dfrac{W}{778}$	1203.0930	1198.8789	973.1920	1032.8986
Loss	D	$Q-B$	82.1813	82.6596	89.7087	100.7501
Loss..............	D'	$Q_a+Q_b+Q_c+Q_d$	82.1813	82.6595	89.7087	100.7501

SUMMARY AND RESULTS.

Quantities.	Symbols.	Formulæ.		1st Trial. Head.	1st Trial. Crank.	2d Trial. Head.	2d Trial. Crank.
Quality of steam entering	X	Per calorimeter.	per cent.	99.41	99.41	97.99	97.99
" " at cut-off	X_1	$100\frac{V_e+V_1}{(M+M_0)C_1}$	"	52.34	51.50	60.986	57.045
" " " release	X_2	$100\frac{V_e+V_2}{(M+M_0)C_2}$	"	63.34	65.26	66.048	69.953
" " " compression	X_3	$100\frac{V_e+V_3}{(M_0+M_x)C_3}$	"	71.75	77.16	94.318	98.338
" " " admission	X_4	$100\frac{V_e+V_4}{M_0C_4}$	"	102.05	102.05	102 00	102.00
" " " "	X_0	Per calorimeter.	"	102.05	102.05	102.00	102.00
" " in exhaust	X_5	$\left(\frac{K+K'}{M-M_x}-S_5\right)+L_5$	"	90.214	90.195	86.328	85.996
Heat lost, admission	a	Q_a+Q	"	42.881	42.905	34 872	37.934
Heat restored, expansion	b	Q_b+Q	"	10.571	12.733	7.068	12.962
Heat rejected, exhaust	c	Q_c+Q	"	23.603	22.270	20.220	17.084
Heat lost, compression	d	Q_d+Q	"	− 2.313	− 1.452	.856	1.008
Heat utilized, work	w	$\frac{W}{778}+Q$	"	5.518	5.518	6.136	6 036
Heat lost, radiation	r	$D+Q$	"	6.394	6.450	8 440	8.887
Ratio, radiation to work		$r+w$	"	1.1588	1.1689	1.3755	1.4723
Ratio, cylinder condensation to work		$a+w$	"	7.7711	7.7757	5.6318	6.2843
Thermodynamic efficiency	E	$(t-t_2)+(461+t)$	"	19.921	19.031	19.715	19.546
Actual efficiency	E_1	$\frac{W}{778}+Q$	"	5.518	5.518	6.136	6.036
Efficiency compared with ideal	E'	E_1+E	"	27.67	28.99	31.12	36.48

the waste by "cylinder-condensation." At release a portion of this heat has been restored to the steam, and aids, to a limited extent, the transformation of heat into work. Ten to fifteen per cent is thus returned by the metal of the cylinder before the exhaust-valve opens and release begins.

At the commencement of the compression the "quality of the steam has risen, by the re-evaporation or separation of the water contained in the charge, to about 75 per cent in the first trial, and to 95 in the second; and here is seen the effect of treatment of the surfaces. The variations between head and crank end, and possibly a part of the greater differences between the same ends on the two days, may be due to differences in the quantities of water collected in various hollows, and in drops adhering to the surfaces. The quality, after compression is completed, is raised by absorption of the heat of compression, and in part by absorption from the cylinder walls, to 102, which indicates some superheating. This result is given, both by the measurements of the indicator-diagrams, and by the use of a calorimeter, ingeniously contrived to sample and test the steam at any desired point in either stroke. Both methods give the same figure. The steam passing into the condenser contains ten per cent moisture before, and about fourteen after, treatment; a result which possibly may indicate that the engine in the first case sent out all its steam as part of a comparatively homogeneous mixture; while in the second trial it may have carried all its water in suspension and uniformly.

The heat utilized by transformation into dynamic energy, for the performance of work, as shown by the indicator, was 5.5 per cent the first day, and 6 per cent the second; showing a gain of nearly 10 per cent in efficiency of engine, and in the quantities of heat, and steam and fuel, consumed. The computed thermodynamic efficiency of the perfect engine, working within the same extreme limits of temperature, would have been 20 per cent, and the real engine thus gave but 28 per cent of the ideal figure the first day, as untreated, and about

35 per cent the second day, after treatment. The economic result of the application of this process, as prescribed by the Author, is thus seen to be, in this instance, a gain of about 10 per cent. Had time for complete application been allowed, the gain would presumably have been correspondingly greater.

The figures, as obtained, on a basis of 2.25 pounds of steam for the ideal engine at efficiency unity, correspond to an expenditure of 40 and of 36 pounds of steam per horse-power per hour for the two cases, respectively: engine as originally operated, and as treated, or to about $4\frac{1}{2}$ and 4 pounds of coal, with economical evaporation. This corresponds to a saving of about three-quarters of a ton of coal per annum per horse-power, of say three dollars a year per horse-power, the interest, at 6 per cent, of fifty dollars per horse-power; while the cost of treatment would amount to but a few cents on engines of considerable power. Should this treatment require repetition, like the cleaning of boilers, it becomes a question, to settle in every case, how thoroughly and how frequently it will pay to go to that expense. The economical aspect of the problem will require careful observation and experimentation.

Professor Peabody has applied Hirn's analysis to the operation of the experimental triple-expansion engines of the Massachusetts Institute of Technology.*

The main dimensions of the engine are as follows:

Diameter of the high-pressure cylinder....	9	inches.
" " " intermediate "	16	"
" " " low-pressure "	24	"
" " " piston-rods..............	$2\frac{3}{16}$	"
Stroke..............................	30	"

Clearance in per cent of the piston-displacements:

High-pressure	cylinder,	head end, 8.83;	crank end,	9.76
Intermediate	"	" 10.4	"	10.9
Low-pressure	"	" 11.25	"	8.84

* Trans. A. S. M. E., 1891; No. ccccIvi.

The following table gives the results:

No. of Trial.	I.	II.	III.	IV.
Duration of test, minutes	60	60	60	60
Total number of revolutions	5299	5228	5173	5148
Revolutions per minute	88.3	87.1	86.2	85.8
Steam consumption during test, lbs.:				
Passing through cylinders	1193	1157	1234	1305
Condensation in h. p. jacket	57	50	29	30
" in 1st receiver-jacket	61	64	69	72
" in inter. jacket	85	92	97	105
" in 2d receiver-jacket	53	50	52	51
" in l. p. jacket	89	76	90	87
Total	1538	1489	1571	1650
Condensing water for test, lbs.	22847	22186	20244	20252
Priming, by calorimeter	0.013	0.012	0.011	0.012
Temperatures, Fahrenheit:				
Condensed steam	95.4	92.1	102.4	105.3
Condensing water, cold	41.9	42.1	43.0	42.8
" " hot	96.1	96.6	106.3	109.6
Pressure of the atmosphere by the barometer, lbs. per sq. in.	14.8	14.8	14.7	14.7
Boiler pressure, lbs. per sq. in., absolute	155.3	155.5	156.9	157.7
Vacuum in condenser, inches of mercury	25.0	25.1	24.1	23.9
Power and economy:				
Heat equivalent of works per stroke:				
High-pressure cylinder	8.44	8.34	9.17	9.52
Intermediate "	7.12	6.95	7.77	8.42
Low-pressure "	9.64	10.06	10.87	11.79
Totals	25.20	25.35	27.81	29.73
Total heat furnished by jackets	27.58	27.02	27.71	28.45
Distribution of work:				
High-pressure cylinder	1.00	1.00	1.00	1.00
Intermediate "	0.84	0.83	0.85	·0.88
Low-pressure "	1.14	1.21	1.19	1.24
Horse-power	104.9	104.2	113.1	120.3
Steam per h. p. per hour	14.65	14.31	13.90	13.73
B. T. U. per h. p. per minute	258.3	252.8	244.6	241.1

The accuracy of the tests is shown by the comparison of the total radiation, as found by collecting and weighing the jacket-condensation when the engine was at rest, and by computing it by aid of the terms depending on the condenser. As has been seen, this is not a check on the interchanges of heat, but the regularity of the results for the intermediate and low-pressure cylinders are considered very satisfactory; as any er-

ror at the high-pressure and intermediate cylinders is carried on through the remainder of the work.

It is noted that the steam becomes drier in passing through the engine, steam-jacketing with steam at boiler-pressure, and is practically dry at release, in both the intermediate and low-pressure cylinders.

It is found that in this case the heat furnished by the jackets is nearly as much as is converted into work, with the cut-off at $\frac{1}{4}$, and is even larger for the cut-off at $\frac{1}{8}$. The heat lost by radiation is one-third as great as that transformed into work. These results are remarkably good, even for the triple-expansion engine, fourteen pounds of steam per horse-power per hour having been rarely reported for even very large engines.

208. Measurements of Gross and Net Power are commonly made by means of the indicator and either the absorbing or the transmitting dynamometer, the former giving the gross or indicated power, the latter showing the amount of power applied by the engine to a brake or to its special purposes, and capable of doing useful work. The efficiency of the engine as a machine, also, is thus determinable, and is measured by the ratio of the dynamometric to the indicated power.

The Transmitting Dynamometer consists of a set of pulleys so arranged that they may be placed between the prime motor and the machinery to be driven by it, while the effort is measured by, usually, a set of springs interposed between the receiving and the delivering pulleys. The magnitude of this effort is often, perhaps generally, automatically recorded on a travelling ribbon or strip of paper, and the speed of the machine is observed. The product of the effort into the velocity of the point at which it is measured is the measure of the work done in the unit of time and of the power expended.

209. The Prony Brake, the Absorbing Dynamometer, or the Dynamometric Brake, has many forms. A simple form of dynamometric brake for small powers is illustrated in Fig. 215. A is the shaft of the motor of which the power is to be determined; B is the pulley or drum on which the brake-blocks are

changed by the bolts *B*, *B*. The lugs *D*, *D* limit the movement of the beam *E*, which is counterbalanced by a fixed

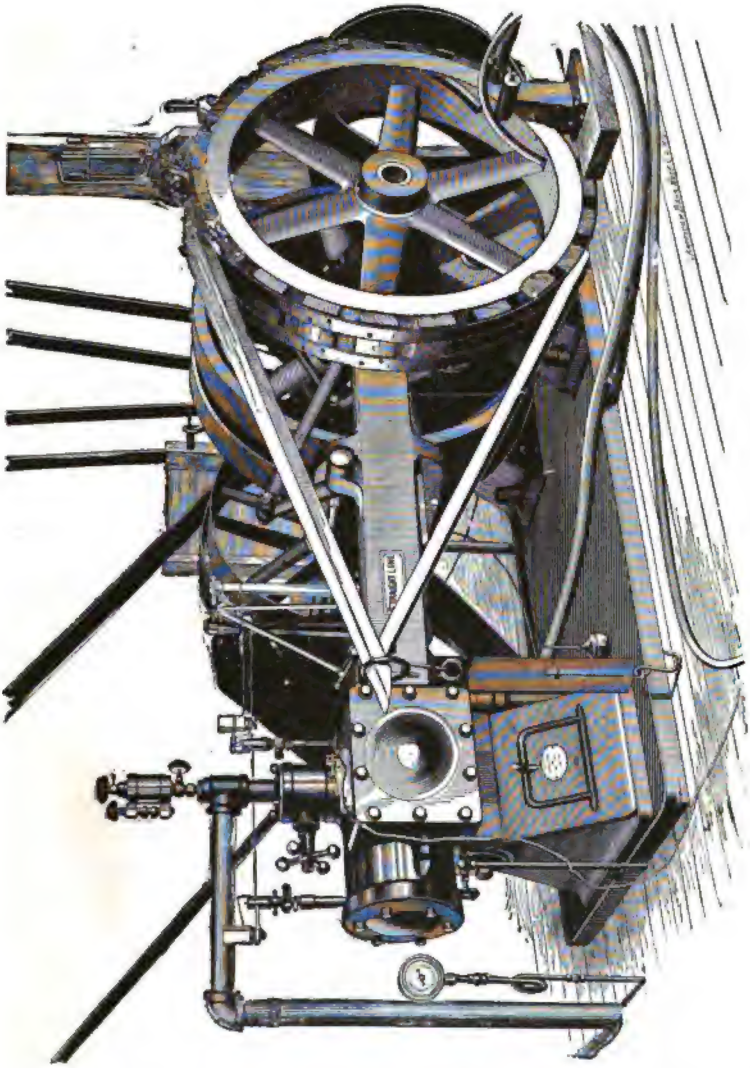

FIG. 214.—FITTING UP FOR TRIAL.

poise at *F*, and the weights equilibrating the effort of the motor are applied at *G*. See also Fig. 214.

Basswood and poplar are excellent woods for use in the rubbing parts of the dynamometric brake; but any wood will work well if properly handled. The soft are usually found better than the hard woods, but white ash and maple are good End-grain is often preferred for rubbing surfaces. The wood may be secured either to the pulley or to the strap of the brake. Where to be much or continuously used under heavy wear, it is perhaps better to put the blocks on the wheel.

The dynamometric power may be ascertained by means of a *rope-brake* upon the fly-wheel of the engine. Two ropes are

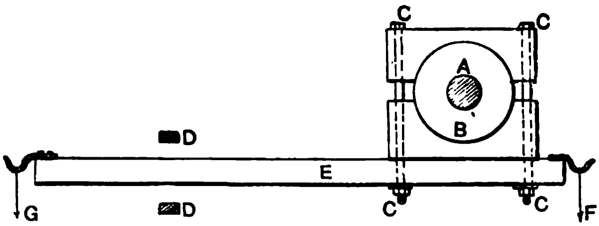

FIG. 215.—DYNAMOMETRIC BRAKE.

used for each wheel, kept at proper distances apart on the wheel by means of transverse wooden distance-pieces. The dead-load is usually applied by means of weights, and the back tension necessary by means of a spring-balance. The spring-balance tension is deducted from the dead-load applied. This brake is found to work perfectly satisfactorily. If any metal be used for attaching the wooden cross-pieces to the ropes, it must not rub against the rim of the wheel; if this happens, the metal becomes hot, and is liable to burn the rope.

If it is assumed that L = length (effective) of the arm of the dynamometer in feet, W = weight (unbalanced) suspended on the arm in pounds, N = number of revolutions per minute, the horse-power will be

$$\text{D. H. P.} = \frac{2\pi WLN}{33000} = 0.0001904\,WLN.$$

It is not unusual to make the effective value of r equal to $\frac{33}{2\pi}$; so that the circumference described being 33 feet, the

power is at once computed by multiplying the weight and number of revolutions of the shaft, *W* and *N*, together, and dividing by one thousand to get the horse-power.

A form of this brake which the Author has most frequently employed, and with satisfaction, includes a brake-wheel, or pulley, which is keyed on the engine-shaft, and is sufficiently strong to sustain safely the maximum load anticipated (Fig. 214).* The rim of this pulley is turned flat and smooth, and fitted with a flexible brake-strap of wrought-iron or other suitable material, which may be adjusted to such a tension as will enable it to control the engine at maximum power. In this case the rim is trough-shaped in section, flanges extending inward toward the shaft to a sufficient depth to permit the retention in the circular trough so formed of a stream of water which is used to keep the pulley cool, and to carry away the heat produced by transformation of mechanical energy, a method first adopted by Mr. Halpin and Professor Sweet. The two ends of the brake-strap are united by a right and left-hand screw, in such manner that they may be drawn together and the strap set up to any desired degree of tension. The brake-arms consist of two beams of wood, forming a < frame, and secured to the strap at the upper and lower sides, and at their junction supported by a strut resting on a platform-scale of nice construction and great accuracy. As the engine-shaft revolves, the tendency of the brake-arms to turn is resisted by the scale: and the effort so measured, multiplied by the relative velocity of the engine-shaft and the supported point on the arm, gives a measure of the power expended. Water is supplied to the pulley-rim, by means of a hose, from any convenient source, and the excess is taken away in a similar manner. The centrifugal action of the rotating mass keeps the fluid in place in the pulley-rim, and the eduction-pipe receives the water carried away by it, as the tender of a locomotive scoops water from between the tracks when at high speed. This system permits

* Construction of a Prony Brake; R. H. Thurston; Journal Franklin Institute; April 1886.

efficient lubrication without admixture of grease with the water, and secures a perfection of smoothness and uniformity of rubbing-surfaces unattainable with older forms of brake.

Professor Alden has recently devised a very ingenious automatically self-adjusting brake; and several modifications of this type using a closed channel in the rim are known.*

Fitting of the Engine for a Laboratory Test, whether of efficiency or of capacity, is done well in advance of the trial,

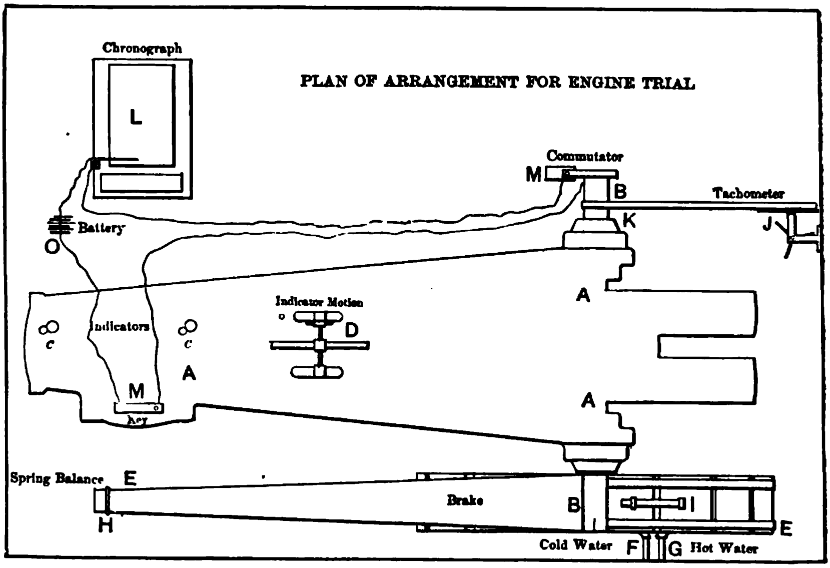

FIG. 216.—FITTING UP THE ENGINE.

and ample time should be taken to see that not only all apparatus, but the engine itself, is in readiness; though, if the intention is, as is sometimes the case, simply to ascertain the condition of the engine as found, no other preparations are permissible than those customary before starting up. A good example of the fitting up of a small high-speed engine is illustrated in Figs. 214, 216, in which *AA* is the engine, *BB* its shaft, *CC* the indicators, *D* the indicator reducing-gear, driven

* See Trans. Am. Soc. M.E.; 1890.

from the cross-head; *EE* is the Prony brake, receiving its cooling-water at *F* and discharging it at *G*, and attached to the *spring-balance* platform at *H*; the screw tightening its strap is at *I*. The speed-indicators were, in this case, of several kinds. Hand instruments of various kinds were used to check the records of the automatic instruments. A "tachometer," *J*, was attached and belted at *K* to the engine-shaft, and afforded a very convenient means of watching the momentary fluctuations due to variations of load, of steam-pressure, and of accidental disturbances. A chronograph at *L* was also attached, connected with a standard clock to beat seconds, and a current was derived from the battery at *O*. A commutator, *M*, was placed on the engine-shaft, making contact at each revolution, and a key, *N*, near the engine, for the purpose of breaking contact.

210. "Experimental Engines," so-called, are illustrated by those not infrequently forming part of the equipment of schools of steam-engineering and of general mechanical engineering, in the construction of which provision is made for such variety and range of adjustment of the conditions of operation as will enable those in charge of them to exhibit all those effects of varying pressures, speeds, ratios of expansion and of valve-setting which affect in any important degree their economical and practical operation. The frontispiece illustrates this type of engine substantially as constructed for the Massachusetts Institute of Technology and, with some changes, for Sibley College, Cornell University, as the form adopted in Europe is illustrated in the frontispiece of Part I. In these instances the value of compounding as an economical expedient may also be shown and, to a certain extent, the principles governing its application illustrated.

In that adopted, as above, by the Author, for use in the Sibley College laboratories, and built by the E. P. Allis Co., a "triple-expansion" engine is formed by placing side by side three complete horizontal engines of, respectively, 9, 16, and 24 inches diameter of cylinder and 30 inches stroke of piston, their frames being united into one, covering a floor-space

about 25 feet by 30. At 100 revolutions and 150 pounds steam-pressure the machine is expected to develop 150 to 200 horse-power. Either engine may be operated alone or in conjunction with either or both its fellows. The valve-gear is that of Corliss, as designed in detail by Mr. Reynolds. It may be worked as usual with such engines; but a twin "wrist-plate" is provided for the exhaust side, allowing complete independence of movement of steam- and exhaust-valves and, by conversion into the "plug-tree" system, the setting of the steam-eccentric with the crank and the adjustment of the point of cut-off anywhere, up to full stroke, and the ratio of expansion from unity to infinity, or either or all. All the cylinders are completely jacketed, and the piping is arranged independently; so that the jacket-water may be separately measured. Any range of pressure up to 12 atmospheres, or more; any speeds up to the limit at which the valve-gear remains efficient; and any ratio of expansion, with or without condensation, may be employed in investigation.

Illustrations of the methods and of the character of the researches conducted with such engines have been already given in earlier parts of this work; as, for example, those of Clark, Hirn, Isherwood, Gately & Kletsch, Delafond, Willans, and others; while the work of Professor Peabody on the first machine above mentioned exhibits the characteristic qualities of this engine.

When these engines are used in the instruction of students, they are given such adjustment as will best illustrate the principles and practice to be taught, and the trial is conducted according to the standard rules usually taken as governing the case. If the usual economical test is to be made, the rules prescribed by the Committee of the American Society of Mechanical Engineers may be followed; if a complete study of the thermodynamics of the engine is proposed, the methods of Hirn's analysis are to be adopted; and if the solutions of problems relating to either the theory of the engine or the effect of stated variations of its working conditions are to be determined, the same applications of precise and refined scientific methods are employed.

CHAPTER VII.

SPECIFICATONS AND CONTRACTS.

211. The Purpose of Specifications and Contracts— for the two papers are really parts of a single legal document —is to insure:

(1) That the exact nature and all details of the work to be done shall be unmistakably expressed in a permanent record.

(2) That the precise manner in which the constructor shall do the work, the kind and amounts of material to be employed, the conditions demanded to be fulfilled by the completed machine, and any later legal or other responsibility for its efficient working and its maintenance, and the time to be given for the completion of the contract, shall all be as exactly and indubitably recorded for future reference.

(3) That the equivalent obligations of the purchaser shall be stated in writing with precision and beyond question, including any assistance agreed to be rendered the builder, either in construction, in location, or in subsequent test-trial and operation, and including all details of the methods of payment and final settlement of accounts, of mutual obligations, and of individual responsibilities. This often also includes a prescription of the method of settlement of any misunderstandings or disagreements that may arise as to either the terms of the contract or unspecified details.

In a word, the contract and its specifications are the exact statement of the terms of the bargain and of all essential details of its fulfilment.

These papers consist of a formal statement of the agreement and of the obligations assumed by both sides, and of specifications of all details, and often with drawings showing

the exact construction of the engine and of its every part. These drawings are made a part of the specification, and the latter is made a part of the contract.

Such papers are always prepared when any important work is undertaken, and should be drawn up, usually, by a legal adviser in consultation with an engineering "expert," both of whom should be experienced and familiar with the kind of contract called for. The documents so prepared are signed in duplicate by both parties to the contract, or by their authorized agents, and each side retains a copy. In some cases other copies are made and signed, and deposited in some safe place for permanent preservation.*

A proposal is commonly presented the intending purchaser by the maker or vendor, which proposal is sometimes also taken, in lieu of a specification, as part of the contract. The following is a form adopted by a successful firm building "drop cut-off engines":

FORM OF PROPOSAL.

We furnish of the very best material and workmanship —— of our standard Automatic Cut-off Engines, having a cylinder of —— inches bore, —— inches stroke, with a —— fly-wheel, —— feet diameter, —— inches face, turned for —— belts, —— inches wide, to weigh about —— pounds. Crank-shaft of best forged and hammered wrought-iron, —— inches diameter, —— feet long. Connecting-rod and all small parts of steel.

Engine to be —— condensing, —— air-pump, —— diameter, —— stroke, exhaust-pipe connections from cylinder to condenser, provided with straight way-valve and branch to atmosphere with valve, so engine can be run condensing or non-condensing, as required. First length of air-pump discharge-pipe with hot-well, injection-pipe and valve with polished iron column and hand-wheel.

Cylinder to be lagged with polished iron, packed with mineral wool, and fitted with Corliss valve-gear, vacuum dash-

* Consult Haupt on Specifications.

pots, and standard regulator; engine to run —— revolutions, and develop economically at that speed, with 75 pounds steam-pressure, cut-off at $\frac{1}{4}$, —— horse-power.

We furnish steam stop-valve, foundation bolts and plates, sight-feed cylinder-lubricator, oil-pump, oil-cups, and necessary wrenches.

We furnish the time of mechanic to erect and put the engine in good running order, ready for your pipe-connections and belting.

The buyer will pay our mechanic's board and travelling expenses and furnish all necessary assistance or common labor, and all masonry and carpenter work, and material for engine foundations and floors, from drawings furnished by us.

He will have engine-foundations ready upon arrival of our mechanic, and if there be any delay on account of foundations not being ready, or if they be imperfectly constructed, waiting for boilers or pipe connections, or anything for which we are not responsible, time of our mechanic during such delay to be charged in account.

Above engine delivered free on board at factory, securely packed for shipment.

We can have the engine ready for delivery in —— days after receipt of order. We guarantee workmanship and materials for one year, provided no disarrangement is caused by carelessness or want of attention of person in charge.

212. **The Nature and Form of the Contract** depends upon the character of the work to be done, and the intended method of making compensation. It is, in all cases, an agreement in writing, by which one party to the bargain agrees to do a certain exactly specified work; while the other agrees to make full, fair, and precisely stated compensation; and, often, penalties agreed to in advance are stated in the contract. Whatever may be required to make the understanding complete, and beyond question or debate, may be embodied in the document, including even advertisements, proposals, preliminary agreements, and all drawings and descriptive specifications.

A good contract is full and definite as to the agreements and obligations of both parties; the compensation must be valid, legal, and exactly stated as to amount and terms of payment; mutual and perfectly voluntary agreement must be obtained, and signature must be of free will. No person below legal age or of unsound mind can take part in a contract. Once made, signed, and delivered, the contract becomes the only legal representative of the bargain, and its interpretation by the courts will be the only binding form of the agreement with both sides. The paper is its own commentary and proof, and every term is taken in its usual acceptation unless otherwise stated by the contract itself. Technical terms are accepted in their generally admitted technical meaning in business. Any obscurity, however, may vitiate the contract. Private and individual original intent or mental reservation have no weight in determining an interpretation.

The following illustrates a form of agreement and contract adopted by the United States Light-house Board:

CONTRACT.

ARTICLES OF AGREEMENT, made and entered into between ....

....

.... the county of ...., and State of .... of the first part, and .... acting for and in behalf of the United States of America, of the second part, witnesseth:

That the party of the first part, in consideration of the matters hereinafter referred to and set out, and of the specifications attached hereto, and the plans forming a part of this contract, covenants and agrees to and with the party of the second part, to furnish all the materials and labor necessary, and to completely construct and deliver, at..... ....

....

....

And the said party of the first part further agrees to forfeit

the sum of twenty-five dollars ($25) per day as liquidated damages for each and every day's delay in completing the work after the said date, said amount to be deducted from any sum due the said party of the first part in the hands of the Light-house Establishment, and to conform in every particular to the stipulations and conditions specified in this contract, and to the specifications and drawings hereto annexed, and which are to be considered as a part of this contract.

The party of the first part agrees to be governed in all matters regarding the work by the party of the second part, or the authorized agent thereof, and that every portion of said work shall be subjected to a rigid inspection by the party of the second part, or his agents appointed therefor, and that this inspection shall be final.

And the party of the second part covenants and agrees to pay the party of the first part, as full consideration and price of and for said iron twin-screw steamer Pansy, ...

....

.... upon receiving satisfactory evidence that the work has been completed and delivered in accordance with the terms of this contract, and to the satisfaction of the authorized agent of the party of the second part: ....

....

....

and as each of the payments is made, possession of the material and labor, which are paid for by said payment, shall pass to and the title thereto shall be vested in the party of the second part.

Provided, however, that in case the party of the second part shall at any time be of opinion that this contract is not duly complied with by the party of the first part, or that it is not in due progress of execution, or that the said party of the first part is irregular or negligent, in such case the said party of the second part shall be authorized to declare this contract forfeited, and thereupon the same shall become null, and the party of the first part shall have no appeal from the opinion and decision aforesaid; and the right to except to or question the same, in any place or under any circumstances whatever,

is hereby released by the party of the first part; but the party of the first part shall remain liable to the party of the second part for the damages occasioned to him by the said non-compliance, irregularity, or negligence.

And it is further stipulated and agreed that no member of Congress shall be admitted to any share or part of this contract or agreement, or to any benefit to arise therefrom; and this contract shall be in all its parts subject to the terms and requisitions of an act of Congress passed on the twenty-first day of April, in the year of our Lord one thousand eight hundred and eight, entitled "An Act concerning public contracts."

And this contract is also expressly understood to be subject to the terms and conditions of the joint resolution of Congress approved April fourteen, one thousand eight hundred and fifty-two, containing a proviso in the following terms, viz.: "Provided nothing herein contained shall be so construed as to authorize any officer or agent of the United States to bind the United States by contract beyond the amount appropriated by Congress, or to sanction any such contract heretofore made."

And it is further understood and agreed that no light-keeper, superintendent, or inspector of lights, engineer officer, nor any other person connected with or engaged in the Light-house Establishment service, shall be allowed to contract for labor or materials, nor to be interested in this contract, nor to any benefit to arise therefrom.

Provided, also, that it is expressly understood and agreed that this contract or any part thereof shall not be sub-let nor assigned; but that it shall be well and truly carried out and fulfilled in good faith by the above recited party of the first part, and that payment on account thereof shall be made to the aforesaid party of the first part, ... heirs, executors, or administrators, or to such person or persons as .... may lawfully authorize, by power of attorney, to receive the same.

And provided further, that this contract shall not be binding upon the United States until it shall have been approved.

And for the true and faithful performance of all and singular the covenants, articles, and agreements hereinbefore particularly set forth, the subscribers hereunto bind themselves, jointly and severally, their and each of their successors, heirs, executors, and administrators.

Thus covenanted, made, and agreed by the parties this .... day of ...., one thousand eight hundred and seventy-............., as witness their hands and seals.

Signed, sealed, and delivered in the presence of—

WITNESSES:

....	........................[SEAL.]
....	.... [SEAL.]
....	.... [SEAL.]
....	.... [SEAL.]
....	.... [SEAL.]

213. **Penalties for Breach of Contract** should always be as explicitly stated and agreed upon in the contract as any other matter of common concern. If left undetermined, suits for damages will be liable to arise, and their settlements in the courts may prove vastly expensive.

214. **The Duties of the Contracting Parties** are, on the one side, to do the work demanded in a satisfactory manner, and on the other side, to make suitable compensation. The duty of each party to the contract is separately defined and described. The "contractor," as the first is called, is bound to perform a specified work in a workmanlike manner, to finish it within a stated time, and to accept payment tendered as prescribed. The purchaser must make full compensation to the extent agreed upon in the specific manner, to promote the work so far as he properly may, to at all times carefully abstrain from either positive or negative action that may embarrass the contractor, always meeting the latter in a friendly and helpful spirit. The work is the contractor's until paid for as prescribed; thus paid for, it becomes the property of the buyer,

who then assumes all risks from which he is not specifically relieved by the contract.

Such penalties are of the nature of a standing debt, and may be similarly held and collected. Failure of either side to meet the requirements of the contract does not necessarily imply freeing the other side from obligations under the contract, unless such failure operates as an interference with the operations of the other. Should no time be stated, damages may still be collected if the work is not finished within what may be settled to be a "reasonable time."

In statements of time, the secular day is taken as beginning and ending at midnight. Actions for damages for breach of contract and enforcement of penalties must, usually, be entered against the delinquent within six years; but the limiting time is not the same in all States. On public works, and sometimes on very important or very extensive private work, a guaranty and bond may be exacted to ensure prompt completion.

The following is an example of such documents as prescribed by the United States Navy Department——

FORM OF BOND WITH CONTRACT.

KNOW all men by these presents, that we,.................
....
....,........
are held and firmly bound unto the United States of America in the full and just sum of..................................
....
dollars, ($............,) lawful money of the United States, to be paid to the said United States, or to its proper agent or attorney duly authorized to receive the same, as liquidated damages; to which payment, well and truly to be made and done, we bind ourselves and every of us, our and every of our heirs, executors, and administrators, in the whole and for the whole, jointly and severally, firmly by these presents.

Sealed with our seals and dated this day of............ anno Domini one thousand eight hundred and...............

The condition of the above obligation is such, that if.....
....
..........heirs, executors, and administrators, do and shall well and truly execute the contract hereto annexed which........ha entered into with..............................
....
for and in behalf of the United States, by which...........
....
conforming in all respects to said contract, the same being hereto annexed, then the foregoing obligation to be void and of none effect; otherwise to remain in full force and virtue in law.

Signed, sealed, and delivered in the presence of—

Witnesses:	
....	..............[L. S.]
....	..............[L. S.]
....	..............[L. S.]
....	..............[L. S.]
....	..............[L. S.]

NOTES.

All signatures of contractors and sureties should have affixed to them seals of wax or wafer, and their first names should be written in full.

The residence of sureties and witnesses should be given.

The bondsmen must qualify in the forms following.

BONDSMEN'S OATHS.

STATE OF................ }
COUNTY OF............. } *ss.*:

................, being duly sworn, deposes and says that he resides at No...
in the State of..................; and that the value of his property, over and above all debts and liabilities incurred by him, is over..
dollars ($............), and that he is fully responsible for

the amount of his obligation in the foregoing bond by him executed.

....

Sworn and subscribed this........day of.........., 18—, before me.

......................[L.S.]

....

STATE OF................ } ss.:
COUNTY OF............ }

................., being duly sworn, deposes and says that he resides at No..
in the State of.................. ; and that the value of his property, over and above all debts and liabilities incurred by him, is over..
dollars ($............), and that he is fully responsible for the amount of his obligation in the foregoing bond by him executed.

....

Sworn and subscribed this.........day of........., 18—, before me.

...................[L. S.]

....

CERTIFICATE OF SOLVENCY.

I CERTIFY that I have made due and diligent personal inquiry as to the ability of the signers of the foregoing bond, and am satisfied that they are good and sufficient and fully responsible for the sum of...................................
dollars ($ ....) each.

....

[OFFICIAL SEAL.] ....

DATE ;...........................

NOTES.

The sureties' certificate of solvency should be signed by an officer of the United States having a seal, and his official seal should be affixed or impressed on the certificate.

When the certifying officer has no seal the fact should be stated.

Each surety must qualify in double the amount of the bond.

215. The Form and Character of the Specification have little or no reference to legal conditions, but determine the extent and the details of the work of construction. It includes the descriptive part of the contract, and this should be so presented that, if possible, the text may be rendered largely superfluous by the introduction of drawings. The character of the materials to be used in every part, the method of shaping them, if not the usual method of the shop, their ultimate size, shape, and finish, all needed detailed instructions, should be carefully given in simple, concise, exact phraseology. The bids of proposing contractors are made on these specifications and their accompanying drawings, and it is thus especially important that they should should present every essential detail. To leave anything to be assumed involves risk.

Where heavy machinery is to be contracted for, complete sets of general and detail working drawings should form parts of the specifications. They should be as complete as if made for use in the shops.

216. Engine Specifications are gradually coming to take standard forms, and to represent more and more completely and thoroughly the best practice of the day. They should always be carefully drawn by the designing engineer, and should be by him laid before the employer and intending purchaser, with the proposed forms of advertisements, agreements, proposal-blanks, and contracts, for his study, and for discussion and modification, if thought necessary, before their publication or use. A good specification of this class includes a statement of the type and power of engine to be called for, the steam-pressure adopted, speed of piston and of rotation proposed, all details of its construction and of its connection with its work, the efficiency, economy, nicety of regulation, and every other essential fact relating to its structure and operation; all stipulations in regard to its trial, its acceptance, and its later operation.

217. The Quality of Materials and their nature in other respects are often made the subject of a distinct specification.

Thus, a specification for iron for car-axles, which would be equally suitable for shafts and other parts of the steam-engine, is made for railroad work in the following form:

Material.—To be made from clean hand-picked No. 1 wrought-iron scrap, free from steel pieces.

Manufacture.—Scrap to be piled in piles of about 200 lbs., heated, and well hammered into blooms. These are to be reheated and hammered into slabs; the slabs to be heated and cemented under the hammer into a billet of rounded section. Each end of such billet to be heated in turn and hammered into finished shape. This gives 4 heats and 4 hammerings from the scrap to the finished axle.

Workmanship.—Axles must be forged closely to dimensions ordered, and cut exactly to length. The makers' name to be stamped upon them plainly.

Working Test.—The metal of finished axle to show tensile strength of 50,000 lbs., with acid test to show close weld and close uniform fibre.

Or as follows:

Material and Manufacture.—To be made of new iron, from good neutral iron, laid in faggots of not more than nine bars; these faggots to be heated and cemented together under the hammer into a rounded billet; each end to be heated and worked into finished shape.

Balance of specifications same as for scrap axle.

In many cases we meet with some such specification as this:

Quality of Iron.—The iron subject to tension shall be tough, ductile, and of uniform quality, capable of sustaining not less than 50,000 lbs. per square inch of sectional area, to have an elastic limit of not less than 26,000 lbs. per square inch; the reduction of breaking area shall average 25 per cent, and the elongation of the bar before rupture to be at least 15 per cent, and when cold a round bar $1\frac{1}{2}$ inches in diameter must bend through 180° without sign of fracture.

Iron subject to compressive strain must be tough, fibrous,

uniform in quality, and have an elastic limit of not less than 25,000 lbs. per square inch.

All cast iron in important details must be good, tough metal, of such quality that a bar 5 feet long, 1 inch square, 4 feet 6 inches between knife-edge supports shall sustain a weight of 500 lbs. on knife edge at centre without breaking.

Parts subject to shearing or bending strains shall be of the best quality of wrought iron.

The following is another example illustrating the requirements of bridge-builders, who have been, on the whole, more generally careful than other constructors. Good steam-engine work is no less exacting, even if it should not be more so:

Quality of Material.—All wrought-iron must be tough, fibrous, uniform in quality, free from flaws, blisters, and cracks, and must have a workmanlike finish. It must be capable of sustaining at least 46,000 lbs. per square inch on a full section of test-piece, with an elastic limit of 23,000 lbs. per square inch.*

All iron to be used in tension or subjected to transverse stress must have a minimum stretch, on a length of 8 inches, of fifteen per cent. measured after breaking.†

All iron to be used in compression must have a minimum stretch of ten per cent on a length of eight inches measured after breaking.

All iron to be used in important parts under tension must be double-rolled after and from the muck-bar (no scrap allowed), and capable of sustaining a stress of 50,000 lbs. per square inch, with an elastic limit of 25,000 lbs. per square inch, and a minimum stretch of twenty per cent measured after breaking in length of 8 inches.

Such tests must be at the expense of the contractor when the requirements of the specifications are not complied with.

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* It will be borne in mind that the strength of iron of good quality varies with the size of the piece. It is rarely assumed in very heavy work that more than 80 per cent of this figure can be relied upon, as iron is found in the market.

† This length is becoming a very widely accepted standard.

All tension wrought-iron, if cut into testing strips 1½ inches in width, must be capable of resisting, without signs of fracture, bending cold by blows of a hammer, until the end of the strips form a right angle with each other, the inner diameter of the curve of bending being not more than twice the thickness of the piece tested. The hammering must be only on the extremities of the specimens, and never where the flexion is taking place. The bending must be stopped when the first crack appears.

All the tension tests are to be made on a standard test-piece of ½ inch to ¾ inch in area of section, planed down on both edges equally, so as to give that area for a length of 8 inches.

All pieces which are to be bent in the manufacture must, in addition to the above requirements, be capable of bending sharply to a right angle, at a working heat, without showing any sign of fracture.

All rivet iron must be tough and soft, and pieces of the full diameter of the rivet must be capable of bending until the sides are in close contact, without showing fracture on the convex side of the curve.

Parts of 4½ inches diameter or less may be rolled iron; but those of greater diameter must be forged.

Parallel rods of locomotives are peculiarly liable to breakage in consequence of their inertia and "fling" when at high speed, and the added stresses due to variations in the wheels which they connect, either as to size or amount of "skidding" or drag. Specifications demand extraordinary care in selection of material. For example, the Pennsylvania Railroad requires that stub billets for such work (which is now usually of steel) must be of such quality that test-pieces, drawn down under the hammer and turned to ⅝ inch diameter and 2 inches length, must bear 85,000 pounds per square inch, and an extension of 15 per cent.; any piece falling below 80,000 will be rejected. The same requirements are made for shafts and axles, and for crank-pins. Merchant bar iron must have a tenacity of 50,000 pounds per square inch (3513 kgs. per sq. cm.) and an elongation of 20 per cent in similar test-pieces; it

is rejected if falling four per cent below that tenacity; the same figures are demanded for coupling links and pins. Chains must stretch 10 per cent, but are allowed a low tenacity.

Springs are used in the valve-gearing of many engines, and sometimes elsewhere. They should be made of spring steel containing about one per cent carbon, and very little of either phosphorus, sulphur, manganese, or silicon. Sulphur or phosphorus to the amount of 0.05 per cent; silicon above 0.10, or manganese in excess of 0.25, or at most 0.50, per cent, should condemn the steel, and specifications for steel for important springs should always so prescribe.

Bronzes for bearings should be carefully specified as to composition.* For heavy journals, and especially at high speeds of rubbing, the following may be specified :

Copper, 80 ; Tin, 10 ; Lead, 10.

If phosphor-bronzes are adopted, from 0.5 to 0.10 per cent phosphorus may be prescribed.

Steel Castings should be specified to be free from blow-holes, cracks, or "cold-shuts," and to have at least the tenacity of the best wrought-iron and not less than 10 per cent elongation.

Feed Cups for steam-cylinders should be "sight-feed" cups, and should be specified to be fitted to use heavy 500° Fahr., fire-test oil, which is usually best mixed with from 10 to 25 per cent. of good "prime" lard oil.

Wood is now never used in engine-construction ; but it may often be required for incidental purposes. The harder and stronger woods are employed only where strength is demanded ; while white pine is almost exclusively used in making patterns and parts not subject to heavy loading.†

* See Chap. XVII.; also Chap. IV. of Friction and Lost Work, etc., by R. H. Thurston; N. Y., J. Wiley & Sons, 1885; also for more complete lists of compositions, Part III., Alloys, etc.; Materials of Engineering; R. H. Thurston; same publishers.

† Consult Materials of Engineering, vol. I.; R. H. Thurston; N. Y., J. Wiley & Son, 1884.

Yellow pine, the wood very generally adopted for framing and to carry loads, should be of the long-leaved variety, cut in Georgia or further South, clear, sound, and straight, and as largely as possible heart wood.

The following are the specifications of the United States Navy Department for materials and workmanship in the steam machinery of Naval vessels:

All castings must be sound and true to form, and before being painted must be well cleaned of sand and scale and all fins and roughness. No imperfect casting or unsound forging will be used if the defect affects the strength, or to a marked degree its sightliness.

All nuts on rough castings are to fit facings raised above the surface. All flanges of castings are to be faced, and those coupled together are to have their edges made fair with each other. The facings of all circular flanges are to be grooved. *All bolt-holes* in permanently fixed parts are to be reamed or drilled fair and true in place, and the bodies of bolts finished to fit them snugly. All threads on bolts and nuts are to be of the United States Navy standard.

All pipes beneath floor-plates are to be connected by forged bolts and nuts of Tobin's metal.

All brasses are to fit loosely between collars of shafting. All brasses or journals are to be properly channeled for the distribution of oil.

Packing for stuffing-boxes to be such as may be approved.

All small pins of working parts are to be well case-hardened.

All materials used in the construction of the machinery are to be of the best quality. The iron castings are to be made of the best pig-iron, not scrap.

Composition castings will be made of new materials. The various compositions will be by weight as follows, viz:

For all journal boxes and guide-gibs, where not otherwise. specified:

Copper, 6 parts;
Tin, 1 part;
Zinc, $\frac{1}{2}$ part.

Tobin's metal:

Copper, 58.22 per cent.;
Tin, 2.30 per cent.;
Zinc, 39.48 per cent.

Naval brass:

Copper, 62 per cent.;
Tin, 1 per cent.;
Zinc, 37 per cent.

For composition not otherwise specified:

Copper, 88 per cent;
Tin, 10 per cent.;
Zinc, 2 per cent.;

Muntz metal to be of the best commercial quality.

Where the drawings call for Muntz metal or naval brass, but not so stated in these specifications, Tobin's metal will be substituted.*

Ornamental brass fittings are to be of good uniform color.

All castings are to be increased in thickness around core-holes. Core-holes are to be tapped and the core-plugs screwed in and locked.

All steel forgings are to be without welds, and to be free from laminations.

All flanges, collars, and off-sets to have well-rounded fillets.

India-rubber valves are to be made of the best Para caoutchouc, with no other ingredients whatever than sulphur and white oxide of zinc; the sulphur is not to exceed 3 per cent. and the oxide of zinc 70 per cent. The India-rubber is not to contain any remanufactured materials, and is to be of a homogeneous character throughout, thoroughly compressed, free from air-holes and all other imperfections.

Any portion of the work, whether partially or entirely completed, found defective, must be removed and satisfactorily replaced without extra charge.

Specifications for lubricants should be drawn with the advice

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* Thurston's "Maximum" Alloys, which are brasses containing from 0.5 to 2 per cent tin, are also allowed. See Mat. Eng'g; vol. III.

of a chemist skilled in that department. The following are samples as adopted by the Pennsylvania Railroad Co. in 1890:

Five different grades of petroleum products will be used.

The materials desired under this specification are the products of the distillation and refining of petroleum unmixed with any other substances, and conforming to the detail specifications as below. Products having very offensive odor, or being mixed with other oils, will not be accepted. Shipments must be made as soon as possible after the order is received. All shipments received at any place on or after October 1st must show the proper cold test, and all received on or after May 1st must show the proper flash point, and will be rejected if they fail, even though the order did not call for winter and summer oil, respectively, unless it can be shown that the shipments have been more than a week in transit. No preliminary examination of samples will be required, but a limited amount of special preliminary examination will be made on the request of the purchasing agent for use of parties desiring the information. When a shipment is received, a single sample will be taken at random and subjected to test, and the shipment will be accepted or rejected on this sample. If rejected, it will be returned at the shipper's expense.

The following detail specifications will be enforced;

150° *fire-test oil.*

This grade of oil will not be accepted if sample:

1. Is not "water white" in color.
2. Flashes below 130° Fahrenheit.
3. Burns below 151° Fahrenheit.
4. Is cloudy, or shipment has cloudy barrels when received, from the presence of glue or suspended matter.
5. Becomes opaque or shows cloud when the sample has been 10 minutes at a temperature of 0° Fahrenheit.

The flashing and burning points are determined by heating the oil in an open vessel, not less than 12° per minute, and applying the test flame every 7° beginning at 123° Fahrenheit. The cold test may be conveniently made by having an ounce

of the oil, in a four-ounce sample bottle, with a thermometer suspended in the oil, and exposing this to a freezing-mixture of ice and salt. It is advisable to stir with the thermometer while the oil is cooling. The oil must remain transparent in the freezing-mixture ten minutes after it has cooled to zero.

300° *fire-test oil.*

This grade of oil will not be accepted if sample:

1. Is not "water white" in color.
2. Flashes below 249° Fahrenheit.
3. Burns below 298° Fahrenheit.
4. Is cloudy, or shipment has cloudy barrels when received, from the presence of glue or suspended matter.
5. Becomes opaque or shows cloud when the sample has been 10 minutes at a temperature of 32° Fahrenheit.

The flashing and burning points are determined the same as for 150° fire-test oil, except that the oil is heated 15° per minute, test flame being applied first at 242° Fahrenheit. The cold test is made the same as above, except that ice and water are used.

Paraffine Oil.

This grade of oil will not be accepted if the sample:

1. Is other than pale lemon color.
2. Flashes below 249° Fahrenheit.
3. Shows viscosity less than 40 seconds or more than 65 seconds when tested as described under "Well Oil" at 100° Fahrenheit throughout the year.
4. Has gravity at 60° Fahrenheit, below 24° Beaumé, or above 29° Beaumé.

From October 1st to May 1st has a cold test of above 10° Fahrenheit.

Fresh Ferry Oil may be used interchangeably with paraffine oil, but viscosity and cold test must be same as for well oil, and gravity at 60° Fahrenheit from 33° to 35° Beaumé. It is not expected that the Ferry Oil can be used in winter.

The flashing point is determined same as for 300° fire-test

oil. The cold test is determined as follows: A couple of ounces of oil is put in a four-ounce sample bottle, and a thermometer placed in it. The oil is then frozen, a freezing-mixture of ice and salt being used if necessary. When the oil has become hard, the bottle is removed from the freezing-mixture and the frozen oil allowed to soften, being stirred and thoroughly mixed at the same time by means of the thermometer, until the mass will run from one end of the bottle to the other. The reading of the thermometer, when this is the case, is regarded as the cold test of the oil.

Well Oil.

This grade of oil will not be accepted if the sample:

1. Flashes, from May 1st to October 1st, below 249° Fahrenheit, or from October 1st to May 1st, below 200° Fahrenheit.

2. Has a gravity at 60° Fahrenheit, below 28° Beaumé, or above 30°.

3. From October 1st to May 1st has a cold test above 10° Fahrenheit.

4. Shows any precipitation in 10 minutes when 5 cubic centimeters are mixed with 95 cubic centimeters of 88° gasoline.

5. Shows a viscosity less than 55 seconds, or more than 100 seconds, when tested as described below. *

* The viscosity test is made as follows: A 100 cubic centimeter pipette of the long bulb form, is regraduated to hold just 100 cubic centimeters to the bottom of the bulb. The size of the aperture at the bottom is then made such that 100 cubic centimeters of water at 100° Fahrenheit will run out the pipette down to the bottom of the bulb in 34 seconds. Pipettes with bulbs varying from $1\frac{3}{8}$ inches to $1\frac{1}{2}$ inches in diameter outside, and about $4\frac{1}{2}$ inches long, give almost exactly the same results, provided the aperture at the bottom is the proper size. The pipette being obtained, the oil sample is heated to the required temperature, care being taken to have it uniformly heated, and then is drawn up into the pipette to the proper mark. The time occupied by the oil in running out, down to the bottom of the bulb, gives the test figures. A stop watch is convenient, but not essential in making the test. The temperature of the room affects the test a little. The limiting figures were obtained in a room at from 70° to 80° Fahrenheit. It will not usually be possible to make duplicate tests without readjustment of the temperature of the oil.

From October 1st to May 1st, test must be made at 100° Fahrenheit, and from May 1st to October 1st at 110° Fahrenheit.

For summer oil the flashing point is determined the same as for paraffine oil; and for winter oil the same, except that the test flame is applied first at 193° Fahrenheit. The cold test is made the same as for paraffine oil.

The precipitation test is to exclude tarry and suspended matter. It is easiest made by putting 5 cubic centimeters of the oil in a 100 cubic centimeter graduate, then filling to the mark with gasoline, and thoroughly shaking.

500° *Fire-Test Oil.*

This grade of oil will not be accepted if sample:

1. Flashes below 445° Fahrenheit.
2. Shows precipitation with gasoline when tested as described for well oil.

The flashing point is determined the same as for well oil, except that the test flame is applied first at 438° Fahrenheit.

Animal Oils.

Two grades of *Lard Oil* known in market as "Extra" and "Extra No. 1" will be used; the former principally for burning, and the latter as a lubricant.

The material desired under this specification is oil from the lard of corn-fed hogs, unmixed with other oils, and containing the least possible amount of free acid. Also from October 1st to May 1st it should show a cold test not higher than 40° Fahrenheit. Oil from lard of "mast-" or distillery-fed hogs does not give good results in service, and should never be sent. Also care should be taken to have the oil made from fresh lard. Old lard gives an oil that does not burn well, and also gums badly as a lubricant. The use of the so-called neatsfoot stock, either alone or as an admixture in making the "Extra No. 1" grade, is not recommended. Neatsfoot oil is used by the Railroad Company when the price will admit, but it is preferred to have it unmixed.

Both grades of oil will be purchased on sample, and shipments must conform to sample. A four-ounce sample is sufficient, and should be sent as directed by the Purchasing Agent. The color of the sample has an influence in the placing of orders. Those lightest in color are regarded as best.

Shipments must be made as soon as possible after the order has been placed. All shipments received at any shop after October 1st will be subjected to cold test and rejected if they fail, unless it can be shown that the shipment has been more than a week in transit.

Shipments of the " Extra" grade will not be accepted, which:

1. Contain admixtures of any other oils.

2. Contain more free acid than is neutralized by 4 cubic centimeters of alkali as described below.

3. Show a cold test above 40° Fahrenheit from October 1st to May 1st.*

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* The cold test of oils is determined as follows : A couple of ounces of oil is put in a four-ounce sample bottle, and a thermometer placed in it. The oil is then frozen, a freezing-mixture of ice and salt being used if necessary. When the oil has become hard, the bottle is removed from the freezing-mixture and the frozen oil allowed to soften, being stirred and thoroughly mixed at the same time by means of the thermometer, until the mass will run from one end of the bottle to the other. The reading of the thermometer, when this is the case, is regarded as the cold test of the oil.

The amount of free acid in oils is determined as follows : Have ready [1] a quantity of 95 per cent. alcohol, to which a few grains of carbonate of soda has been added, thoroughly shaken and allowed to settle, [2] a small amount of turmeric solution, [3] caustic potash solution of such strength that 31¼ cubic centimeters exactly neutralizes 5 cubic centimeters of a solution of sulphuric acid and water, containing 49 milligrams H_2SO_4 per cubic centimeter. Now weigh or measure into any suitable closed vessel,—a four-ounce sample bottle for example,—8.9 grams of the oil to be tested. Add about two ounces of the alcohol, warm to about 150° Fahrenheit, add a few drops of the turmeric solution, and shake thoroughly. The color becomes yellow. Then add from a burette graduated to cubic centimeters the caustic-potash solution, little at a time, with frequent shaking, until the color changes to red, which red color must remain permanent after the last vigorous shaking. The number of cubic centimeters used shows whether the oil stands test or not. In the case of the " Extra " grade, if more than four are used, or in the " Extra No. 1 " grade more than thirty are used, the oil fails.

4. Show coloration when tested with nitrate of silver as described below.*

Shipments of "Extra No. 1" grade will not be accepted, which

1. Contain admixtures of any other oils.

2. Contain more free acid than is neutralized by 30 cubic centimeters of alkali as described below.

3. Show a cold test above 45° Fahrenheit from October 1st to May 1st.

No. 1 Neatsfoot Oil will not be accepted if it contains admixtures of any other oils, nor if it contains more than fifteen (15) per cent of free acid.† From November 1st to April 1st, this grade of oil will not be accepted if it shows a cold test higher than 45° Fahrenheit. Neatsfoot oil will be purchased

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* The nitrate of silver test is as follows: Have ready a solution of nitrate of silver in alcohol and ether, made on the following formula:

Nitrate of Silver,	1 gram.
Alcohol,	200 grams
Ether,	40 "

After the ingredients are mixed and dissolved, allow the solution to stand in the sun or in diffused light until it has become perfectly clear; it is then ready for use and should be kept in a dim place and tightly corked.

Into a 50 cubic centimeter test-tube put 10 cubic centimeters of the oil to be tested (which should have been previously filtered through washed filter paper), and 5 cubic centimeters of the above solution, shake thoroughly, and heat in a vessel of boiling water fifteen minutes, with occasional shaking. Satisfactory oil shows no change of color under this test.

† The amount of free acid in oils may be determined by adding a measured amount of the oil to a couple of ounces of alcohol and then titrating the solution with dilute standard alkali. The details of manipulation, strength of solutions, etc., will be given if desired.

The cold test of oils may be determined as follows: A couple of ounces of oil is put in a four-ounce sample bottle, and a thermometer placed in it. The oil is then frozen, a freezing-mixture of ice and salt being used if necessary. When the oil has become hard, the bottle is removed from the freezing-mixture and the frozen oil allowed to soften, being stirred and thoroughly mixed, at the same time, by means of the thermometer, until the mass will run from one end of the bottle to the other. The reading of the thermometer when this is the case, is regarded as the cold test of the oil.

on sample, and shipments will be required to conform strictly to the sample.

Tallow Oil used for lubrication will not be accepted if it contains admixtures of any other oils, nor if it contains more than 15 per cent. of free acid. From November 1st to April 1st, tallow oil will not be accepted if it shows a cold test higher than 45° Fahrenheit.

Tallow oil will be purchased on sample, and the above specifications will be strictly followed. The color of the sample will also be considered, those oils being regarded as best in this respect which are lightest in color.

Tallow for use in locomotive cylinders should contain the least possible amount of free acid, and should at the same time be as free as possible from dirt, *cracklings*, and fibre. In order to secure such tallow the following specifications have been adopted:

1. Tallow which, on inspection, is found to contain dirt or *cracklings* disseminated through it, or in streaks, or which has a layer of dirt or *cracklings* in the bottom of the barrel more than an eighth of an inch thick will be rejected.

2. Tallow containing more than one and a half (1½) per cent. of free acid will be rejected.

3. Tallow containing soap, or other substances not properly belonging to tallow, will be rejected.

To persons furnishing tallow who may not have appliances for determining the amount of free acid in tallow, it may be said, that if the fat is rendered within twelve (12) hours from the time the animal is killed, using a temperature of not more than 225° to 250° Fahrenheit during the rendering, it is believed that the free acid in the tallow will be less than amount specified above. In very warm weather it may be necessary to render the fat in less than twelve (12) hours after the animal is killed.

218. Workmanship is Specified in a contract, as a general rule, as to be "first-class," "of the best quality," or as "thoroughly satisfactory" to the purchaser or his inspector; but these phrases are all too vague to induce any definite char-

acter of construction or finish. Only careful inspection can secure good work, however stringent the specification; and it usually comes simply to an agreement to permit the inspector to pass what his judgment approves. It is not by any means always economical to specify or to exact "first-class" workmanship in all parts or in every machine. It often happens that the extra cost of fine work, of perfect fitting, and of fine finish, is far more than commensurate with the advantages gained. In steam-engine work generally, however, and in high-speed engines, invariably, the best of workmanship is needed to give permanent and satisfactory economical performance and freedom from excessive costs of repair.

As a rule, the specifications will provide that, without exception, the whole of the construction must be first-class work, and in strict accordance with the drawings and specifications. In the case of sub-contractors, the specifications are fully binding on them in every respect, and free access and information is to be given by them for thorough inspection of material and workmanship, and all required test-pieces, etc., properly shaped, are to be provided as may be requested, without charge. All shipments of material not properly inspected and passed are at the risk of the contractor. No alterations are to be made unless authorized.

219. Design-specifications are included in a contract whenever the machinery is to be made from new designs which have been prepared in advance by the consulting or designing engineer, or are to be made by the builder. They should be very complete and exact. If already made, the drawings always accompany the specifications. It is customary for the departments of public works of cities, for example, to advertise for proposals to furnish machinery to such design-specifications, the latter being accessible to the bidder in the office of the department or of its engineer. When the contractor is to make his own drawings, it is often sufficient to give the general specifications and to permit him to arrange and design details to suit himself, relying upon trial and performance, when completed, to ensure satisfactory work.

220. Operation and Maintenance, the machine being completed, may often be required of the contractor or builder for a fair and limited period of time; but it is oftener thought best to leave the machinery in the hands of the buyer, to operate and maintain, the builder being held liable, under the contract, to make good any deficiencies, and to make at his own expense all extraordinary repairs or injury traceable to defective material or workmanship under usual conditions of operation. When an engine is built to displace an older type, it is sometimes operated by the contractor to exhibit, for a time, its best work. Payment is sometimes, under such circumstances, made dependent upon the amount of saving effected.

221. Sample Specifications may usually be obtained of any builder of important or of standard types and makes of steam or other machinery. These are often very complete. Navy specifications for engines and boilers of large ships are sometimes printed in book-form, and cover nearly 100 pages, 12mo, apart from the drawings.

A QUADRUPLE-EXPANSION ENGINE for U. S. N. torpedo-boats with twin screws, designed by the Bureau of Steam Engineering, has specifications of which the following is a brief abstract,* the proposed speed of boat being 24 knots (27.64 miles, nearly), per hour:—

General Description.—Engines to be right and left in one water-tight compartment, vertical-direct-acting, inverted cylinders, quadruple expansion. Cylinders to be 11½, 16, 21½ and 3c inches diameter, 16 inches stroke of piston. Collective power of both engines, 1800 I.H.P. at 412 revolutions per minute. The high-pressure cylinder will be forward, and the low-pressure aft. The main valves will be of the piston type for the high-pressure and intermediate-cylinders, and slide valves for the low-pressure worked by Stephenson slotted links. There will be one piston valve for each high-pressure cylinder, and two for each intermediate-pressure cylinder, and one double-

* Specifications: Bureau of Steam Engineering, 1891.

ported slide valve for each low-pressure cylinder. The framing of the engines will consist of forged-steel columns, trussed by forged-steel stays. The engine bed-plates will be of plate steel, supported on mild steel keelson-plates built in the vessel. The crank-shaft will be made in one section and will be hollow. The working parts generally will be forged of mild open-hearth steel, the piston and connecting rods to be oil tempered.

The condenser will be made entirely of composition and rolled brass, and will have a cooling surface of about 1,379 square feet, measured on the outside of the tubes, the water passing through the tubes. There will be one single-acting air pump for each engine, worked off the main shaft. The circulating-pump will be centrifugal, worked independently. The propellers will be of manganese bronze, or approved equivalent metal.

There will be two water-tube boilers, placed in two watertight compartments. They will be constructed for a working pressure of 250 pounds per square inch.

There will be an auxiliary feed-pump in each fire-room.

The main feed-pumps will connect with the main feed-pipes, and each set will have a capacity sufficient to supply the amount of water necessary at full power under forced draught. The main feed-pumps will be driven off the main shaft.

There will be two smoke-pipes.

The forced-draught system will consist of one blower in each, which will discharge into the fire-rooms. Bulkheads and air-locks will be fitted to make the fire-rooms air-tight.

There will be evaporators and distillers, and such other auxiliary or supplementary machinery, tools, instruments, or apparatus as are described in the detailed specifications or shown on the drawings. The weight of all machinery and boilers, including auxiliaries, and including water in the boilers, condensers, and pipes, but not including stores, spare parts, heaters, steering-gear, and capstans, will not exceed 53 tons.

Cylinder-castings of best quality of cast-iron, as hard as can

be properly worked. The casings will include the valve-chests, steam ports and passages, and, except in the case of the low-pressure cylinders, the various brackets to which the cylinder supports will be attached.

All the cylinder casings will be bolted together by body-bound forged-steel bolts.

Cylinder-casings will be of close-grained cast-iron as hard as can be properly worked. Each will have one piston-valve.

Cylinder-covers will be of cast-steel, ½ inch thick, and stiffened by ½-inch ribs. The lower heads will be faced and bored for piston-rod stuffing-boxes and for the supports for the upper end of the crosshead guides. The lower head of the low-pressure will have, in addition, facings for the columns. They will be so formed as to leave as little clearance as practicable. The covers of the high-pressure cylinder will be secured to the cylinder-casing by twelve ¾-inch steel bolts, the covers of the first intermediate by sixteen ¾-inch, the covers of the second intermediate by twenty ¾-inch, the upper cover of the low-pressure cylinder by twenty-six ¾-inch, and the lower by twenty-four ⅞-inch.

Holes will be drilled and tapped for jack-bolts and eye-bolts.

Care will be taken that the clearances in the cylinders are made no larger than absolutely necessary. After the engines are set up in place and connected, the volume of the clearance at each end of each cylinder will be carefully measured by filling the space with water or oil, and the result plainly marked on some conspicuous part of the cylinder-casing. Marks will also be made on the crosshead guides, showing the position of the pistons when the clearances were measured.

The valve-chest of each high-pressure cylinder will be fitted for one 5¾-inch piston-valve, each first intermediate-pressure for two 5¾-inch piston-valves, each second intermediate-pressure cylinder with two 7½-inch piston-valves, and each low-pressure cylinder for one double-ported slide-valve.

The steam and exhaust passages must be thoroughly

cleaned out and pickled, and care taken that the passages are nowhere contracted to less than the specified areas.

Each intermediate and low-pressure valve-chest will have a $1\frac{1}{2}$-inch adjustable-spring safety-valve of approved pattern.

All valve-chests will also be fitted with suitable composition drain-cocks or valves that may be operated from the working-platform, the valves to discharge through pipes into the bilge and feed-tank, with the necessary valves for directing the water to either.

Valve-chest covers will be made of cast-steel. They will be finished all over on the outside.

Piston-valves will be of cast-iron, made with two heads, separated by a wrought-steel distance-piece, and each head packed by a single cast-iron ring.

Valve-stems will be of forged steel. The lower end of the high and low pressure stems will be formed into a jaw for the link-block. The lower end of each intermediate-pressure stem will be secured to its crosshead by a collar-nut above and below the crosshead, the nuts and valve-stems being kept from turning by clamps and set-screws. The thread on each stem must be sufficiently long to allow a reasonable latitude of adjustment.

A split-pin will be put through the valve-stems to keep the nuts from coming off.

There will be a $1\frac{1}{2}$-inch adjustable spring relief-valve on each end of each high-pressure and each intermediate, and one 2 inches diameter on each end of each low-pressure cylinder. The valves and their casings will be of composition. Pipes will lead from the relief-valves to the bilge with easily broken joints.

Each cylinder will be fitted with a 1-inch asbestos-packed drain-cock, placed so as to drain the cylinder thoroughly. The cocks must be perfectly tight, without undue friction. The drain-cocks of each cylinder of each engine will be worked by a separate lever at the working-platform. All the drain-cocks of each engine will discharge into a pipe leading to

the fresh-water side of the condenser, with a branch to the bilge.

Each engine will have a $4\frac{3}{8}$-inch stop and approved throttle-valve placed in the same casing, the throttle-valve being next the high-pressure cylinder casing and bolted to it. Each throttle-valve will be worked by lever at the working-platform.

Pistons will be made of cast or forged steel, and will be dished. Each will have one packing-ring, $1\frac{5}{8}$ inches wide and $\frac{5}{8}$ inch thick, of hard cast-iron, cut obliquely, and tongued. The packing-rings will be sprung in without followers. There will be sufficient clearance between the piston and cylinder to allow for difference of expansion. Each piston must be carefully gauged, and care taken that the clearance between the piston and cylinder head and cover is as called for on the drawings. When completed the pistons must be carefully weighed, and no excess of weight will be allowed over that due to the dimensions shown in the drawings.

The piston-rods will be of forged steel, oil tempered, $2\frac{1}{8}$ inches diameter, for all cylinders. They will be turned to fit the pistons, with collars, as shown, and fitted each with a forged-steel nut at the piston end, secured by a split-pin. The lower ends will have the crossheads forged on. The parallel parts will be smoothly and accurately turned. Each rod will have, at its seating in the piston, a collar $2\frac{5}{16}$ inches diameter for all cylinders and $\frac{5}{8}$ inch thick, well filleted, and recessed in the piston as shown. The piston-rods will be kept from turning in the pistons by stop-pins.

The crossheads will be of forged steel. The bearing for the connecting-rod wrist-pin will be formed in the crosshead, with cap fitted with brasses, as shown in the drawing. The pins will be $2\frac{1}{2}$ inches in diameter by $4\frac{1}{2}$ inches long in bearing, and secured firmly in the connecting-rod forked end. Each crosshead will be forged on the piston-rod, and will work on a hollow composition slipper-guide fitted for water circulation. There will be no gibs or liners on the crosshead. The sliding surface for all cylinders will be $7\frac{1}{2}$ by $9\frac{1}{2}$ inches.

The connecting-rods with their caps and bolts will be of

forged steel finished all over. They will be 38 incnes long between centres. They will be forked at the upper end with crosshead-pin secured as before specified. The body of the rod will be 2¼ inches diameter at the crosshead end and 2½ inches diameter at the crank-pin end. The crank-pin end will be fitted with brasses lined with white metal. There will be a 1¼-inch hole through the centre of the rod, forming an oil way to the crank-pin bearing.

Composition distance pieces will be fitted between the crank-pin brasses. They will be so fitted as to be removable without taking out the cap-bolts, and will be channeled so as to be easily reduced when taking up lost motion.

The brasses for each crank-shaft bearing will be lined with approved white metal, fitted in dovetailed recesses and hammered in place; they will be fitted with ample oil-channels and faced at ends, and to fit the bed-plates, and accurately bored to fit the journals of shaft. The caps will be of cast or forged steel, fitted with white metal, with lips to match the jaws. Each cap will have an oval hand-hole for the purpose of feeling the journal. This hand-hole will have a cover, with handle—the lower part of the cover being formed into a perforated tallow-box, reaching to within a quarter of an inch of the journal. The caps for all crank-shaft bearings will be secured by two stud-bolts of forged-steel 1⅝ inches in diameter; the part beyond the nuts will be fitted with a split-pin.

After the engines are secured in the vessel, the brasses and caps will be bored out in place to perfect alignment, if required. They will also be tried on their shafts, and any defects made good by scraping to a proper bearing.

The brasses will be so fitted that the only bearing of the journals will be on the surface of the white metal.

All working parts of the machinery will be fitted with approved and efficient lubricators, each with a sufficient oil capacity for four hours' running. Each lubricator to be fitted with a tube leading to the wipers on the moving parts, or tubes in the bearings and guides. Each tube from the

lubricators will be fitted with a valve adjustment, and a sight-feed with a well-protected glass tube.

Unions will be fitted where necessary, so that the oil-pipes may be quickly taken down and cleaned, and each pipe will be connected to the bearings by a union-joint.

As far as possible all the oil for the moving parts of each engine, except main bearing, will be supplied from one oil-box on the cylinder, with separate valve, sight-feed, and pipe for each part to be oiled. All working parts for which oil-cups are not specified or shown in drawings will have oiling-gear of approved design, such that they can be oiled without slowing. All the oiling of each auxiliary engine will be done by one oil-box where practicable, All fixed oil-cups will have hinged covers, with stops to prevent being opened too far. Moving oil-cups, where necessary, will have removable covers.

All castings must be sound and true to form, and before being painted must be well cleaned of sand and scale, and all fins and roughness removed. No imperfect casting or unsound forging will be used if the defect affects the strength or to a marked degree its sightliness.

All bolt-holes in permanently fixed parts will be reamed or drilled fair and true in place, and the bodies of bolts finished to fit them snugly.

All pipes beneath floor-plates will be connected by forged bolts and nuts of rolled manganese or Tobin bronze.

All small pins of working parts will be well case-hardened.

All steel joint-pins of valve-gear will be hardened and ground to true cylindrical surfaces.

All materials used in the construction of the machinery will be of the best quality. The iron castings will be made of the best pig-iron, not scrap, except where otherwise directed.

Composition castings will be made of new materials. The various compositions will be by weight as follows:

For all journal-boxes and guide-gibs, where not otherwise specified:

Copper 6, tin 1, and zinc $\frac{1}{4}$ parts.

Naval brass (or Thurston's "Kalchoids"):

Copper 62, tin 1, and zinc 37 per cent.

For composition not otherwise specified:

Copper 88, tin 10, and zinc 2 per cent.

Muntz metal will be of the best commercial quality.

Anti-friction metal will be of approved kind.

Ornamental brass fittings will be of good, uniform color.

All work will be in every respect of the first quality and executed in a workmanlike and substantial manner.

Any portion of the work, whether partially or entirely completed, found defective, must be removed and satisfactorily replaced without extra charge.

All steel used in the construction of the boilers, and all steel forgings and castings, will be tested in accordance with rules prescribed by the Navy Department.

All boiler and condenser tubes will be tested to 300 pounds pressure per square inch, applied internally, before being put in place.

India-rubber valves, taken at random, must stand a dry-heat test of 270° F. for one hour, and a moist-heat test of 400° F. for three hours, without injury.

Before the boilers are painted or placed in the vessel they will be tested under a pressure of 340 pounds to the square inch above atmospheric pressure. This pressure will be obtained by the application of heat to water within the boilers, the water filling the boilers quite full.*

The steam-pipes and valves, the auxiliary engines, and all fittings and connections subjected to the boiler-pressure will be tested by water-pressure to 340 pounds to the square inch.

The high-pressure cylinders, jackets and valve-chests will be tested by water pressure to 300 pounds to the square inch, the

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* New boilers should always be tested, before they are painted, with the full pressure permitted by law. They should then be painted on the outside with two coats of brown zinc and oil, and when in place the fronts be painted with one coat of black paint. Where pressures are high, a small "sectional" safety-valve is often placed where its opening will give the alarm in advance of the opening of the main valves.

first intermediate-pressure cylinders and connections to 200 pounds, the second intermediate-pressure to 150 pounds, and the low-pressure to 100 pounds. The exhaust side of the low pressure valve-chests will be tested to 30 pounds. The condensers will be tested to 30 pounds.

The pumps, valve-boxes, and air-vessels of the feed, fire, and bilge pumps will be tested to 350 pounds per square inch. The cylinders and condensers will be tested before being placed on board, and must be so placed that all parts may be accessible for examination by the inspector during the tests. All parts will also be tested after being secured on board. No lagging or covering is to be allowed on the cylinders or condensers during the tests.

The following are sample extracts from the specifications of an English (London, Chatham, and Dover) railway, of 1888, and exhibit the proportions of the main parts:

General Conditions.—The engines and tenders are to be made to the dimensions given in this specification, and to the drawings to be supplied by the company's locomotive engineer, except in cases where his consent to an alteration has been first obtained in writing. The quality of the materials to be of the make specified, and where no instructions are given, both materials and workmanship are to be of the very best description. No advantage whatever is to be taken of any omission of detail, or discrepancies that may occur in the drawings or in this specification, as the contractor may obtain a full explanation of any part of the work not sufficiently shown or understood. The engines and tenders must be finished, in every respect, in the most complete manner, and to the entire satisfaction of the company's locomotive engineer, who shall be at liberty to inspect, either personally or by deputy, the work during its progress, and to reject any defective or unsuitable materials or workmanship.

In case of any dispute arising, either during the progress of the contract or at its termination, the decision of the locomotive engineer of the company is to be taken as final and binding in every respect.

The engines and tenders are to be delivered by the builders, free of charge, fit and ready for work, by the time stated in the tender attached to this specification, and if not so delivered the contractor shall forfeit to the aforesaid company the sum of twenty pounds (£20) per week for each engine and tender in arrear. Prior to payment, each engine and tender will be required to run three thousand miles (consecutively), without showing any defect in materials or workmanship, and the builders will be held responsible for all defects that may appear (accidents excepted) until they have run that distance.

All royalties and patent rights must be paid by the contractors.

Quality of materials. Brass and Gun Metal.—Where "brass" is specified it must be good, tough metal. Gun-metal must be composed of 5 parts of copper to 1 part of tin.

White Metal.—This must be composed of—

Tin....................................	16 parts.
Antimony............................	2 "
Copper...............................	1½ "

Other materials to be obtained of the manufacture to be hereinafter specified, unless the consent of the company's locomotive engineer, in writing, be first obtained to an alteration.

Boiler.—Barrel, dome, fire-box, casing, and smoke-box tube-plate, and all angle-irons, rivets, and stays, to be made of Lowmoor, Bowling, Taylor's, or Cooper's (best Yorkshire) iron. Barrel to be telescopic, and made in two plates, the circumferential seams to be single riveted, and the longitudinal seams to be butt-jointed with inside and outside strips, double-riveted. Tube-plate to be attached to barrel by a ring of angle-iron, bored, faced, turned on edges, and shrunk on, and zig-zag riveted to both. The dome to be in one plate, welded at the seam, and flanged at the bottom to fit barrel, to which it is to be double-riveted, a strengthening liner-plate to be placed inside the barrel, round the opening for the dome. The top to have an angle-iron ring, riveted to it, and to be fitted with

a strong wrought-iron cover; the cover and angle-iron must be accurately faced so as to make a perfectly steam-tight joint.

Fire-box shell to be made as shown, the sides and top in one plate, the front or throat plate flanged forward and single-riveted to the barrel, and the back-plate to the sides and top as shown. Angle-irons for carrying the sling-stays to be riveted to the top in the position shown.

The manhole to be of wrought-iron, flanged top and bottom and single-riveted to the casing, the top flange to be accurately faced to receive the safety-valves. The boiler to be stayed by six longitudinal stays, screwed into the back-plate of casing, and passing through the smoke-box tube-plate, with nut and washer on either side, the back-plate to be strengthened where the stays pass through by a liner-plate, riveted to it on the inside. The longitudinal stays to be supported in the middle of their length, in the manner shown. The fire-hole to be circular, the ring of the section shown to be Yorkshire iron, riveted to the casing and fire-box plates, flush outside casing. The foundation ring to be Yorkshire iron, the corners of the form shown, carefully riveted so as to be thoroughly tight. Twenty-one brass taper mud-plugs, for washing out, to be placed in the position shown on the drawings.

Dimensions.

Length of barrel	10 ft.	3 in.
Diameter of barrel outside at fire-box end	4 "	3 "
Thickness of barrel-plates	..	$\frac{7}{16}$ "
" " tube-plate	..	$\frac{7}{8}$ "
" " dome-plate	..	$\frac{9}{16}$ "
Length of fire-box shell	5 "	9 "
Breadth " " at bottom outside	3 "	11 "
Depth " " from centre of boiler	5 "	2 "
Thickness " " plates	..	$\frac{1}{2}$ "
Section of foundation-ring 3 in. × $2\frac{3}{4}$ in	..	..
" fire-hole ring 2 in. × $2\frac{1}{2}$ in.	..	..
Diameter of rivets in boiler	..	$\frac{13}{16}$ "
" " foundation-ring	..	$\frac{7}{8}$ "
Height of centre-line of boiler from rail	7 "	2 "

Fire-box.—The fire-box plates to be of copper of the very best quality obtained from Messrs. Everitt & Sons, Grenfell & Sons, Vivian & Sons, or other approved makers. The stays to be made from the very best soft-rolled copper bars by the same maker as the plates. The plates to be annealed, both before and after flanging, and strips cut off must be tested by being doubled cold, without showing any sign of fracture, and also analyzed, and must not contain more than .5 per cent of impurities. The sides and crown to be in one plate, the crown curved as shown, and stayed with eight Yorkshire-iron roof-bars of the section shown, each secured by thirteen studs 1 inch diameter, screwed through the crown-plate into the bar, with nut on the under side of plate. Six of the roof-bars to be connected to the angle-irons on the casing plate by twelve sling-stays of Yorkshire iron. Great care must be taken to bed the ends of the roof-bars accurately on the fire-box plates, and that the sling-stays are of the correct length, bearing on the pins top and bottom.

The tube-plate to be stayed to barrel by six 1-inch copper stays, screwed through the plate into palm-stays riveted to the barrel. The copper stays to be screwed tightly into the fire-box and casing plates, and neatly riveted over at the ends, the thread being turned off the portion between the plates. A brass plug with fusible centre to be inserted in the crown of the fire-box. A brick arch to be built in the fire-box supported on studs in the manner shown. Fire-box back-plate to be dished at fire-hole to meet the ring, and the fire-hole fitted with air-deflector scoop and sliding-doors, in the manner shown on the drawings.

Fire-grate to consist of nineteen wrought-iron fire-bars, and two cast-iron fire-bars of the section shown, supported on two cast-iron comb-bar bearers by four wrought-iron brackets, studded to foundation-ring. Fire-box to be riveted with best Yorkshire-iron rivets.

Dimensions.

Length at top outside	5 ft.	½ in.
" " bottom	5 "	2 "

Breadth at bottom	3 ft. 4 in.
Depth inside	6 " 0 "
Water-space at bottom, all round	3 "
Thickness of plates	$\frac{1}{2}$ "
" " tube-plate	$\frac{13}{16}$ & $\frac{1}{2}$ "
Diameter of fire-hole	1 ft. 3 "
" " rivets	$\frac{13}{16}$ "
" " copper stays	$\frac{7}{8}$ & 1 "

All the plates are to be planed or turned on the edges before being put together. The holes must be drilled or punched slightly countersunk, rhymed out perfectly fair with each other in all plates and angle-irons, drifting will under no circumstances be allowed; care must be taken that the smaller diameters of the holes come together, that all burs are carefully filed off, and that the plates are brought well together before any rivet is put in. All rivets must completely fill the holes, and the heads must be perfectly true and central. Any caulking that may be required must be done with a broad-faced tool, so that the plates may sustain no injury. Before being lagged the boiler is to be tested in the presence of the company's locomotive engineer, or his inspector, to a pressure of 200 lbs. per square inch with water, and afterwards to 160 lbs. per square inch with steam, and it must be perfectly tight under these pressures.

Tubes to be of copper, solid drawn, of either Everitt's Broughton Copper Company's, or other approved make, 9 B.W.G. at the fire-box end tapering to 12 B.W.G. at the smoke-box end. To be secured by a roller tube-expander (great care being taken that the tubes are not cracked), and fixed with ferrules at the fire-box end. Ferrules to be of ferrule steel, and to go into the tubes a tight driving fit. The tubes are to project through the smoke-box tube-plate $\frac{7}{16}$ inch.

Dimensions.

Number	199
Length between tube-plates	10 ft. 7 in.

Diameter outside	..	1¾ in.
" " at smoke-box end for a length of 6 inches...............................	..	1⅞ "
Thickness at fire-box end.........No. 9 B.W.G.	..	..
" " smoke-box end.....No. 12 B.W.G.	..	..
Distance apart of centres.............. .about	..	2½ "

Frames.—Inside frames and front buffer-plate to be of Bessemer steel, solid rolled, of Cammell & Co.'s, John Brown & Co.'s, Bolton Iron and Steel Co.'s manufacture, or other approved makers. Each plate must have the brand of the manufacturer legibly stamped on its outer side. The plates are to be planed all over on the inner side, and the outer side must be finished with a good smooth surface. All holes to be marked from one template, and drilled and rhymed out to the exact size given.

The frames to be set in and thoroughly well stayed together by the buffer-plate, and with plates and angle-irons at the leading end in the manner shown on drawings, the front foot-plate to be thinned at the edges as shown. A plate is to be placed horizontally under the cylinders to carry the bogie pin, and must be firmly bolted to angle-irons on the frames. A transverse stay arranged to carry the back ends of motion bars and the intermediate spindle-guides, and a vertical stay in front of the fire-box casing, must be placed in the positions shown. Over the trailing-axle a horizontal flanged stay is to be securely bolted to the frames, and at the hind end of frames a cast-iron foot-plate, arranged for the tender couplings, is to be placed. All these stay-plates and angle-irons to be of BB Staffordshire iron. The casting and the transverse stays must be securely fastened to the frames, the former by turned bolts, and the latter by cold-turned rivets of Lowmoor iron. The rubbing-pieces for tender buffers to be well case-hardened. When finished the frames must be perfectly true and square in all directions. The foot-plate to be of steel, of the same make as the frames, and the rivets to be countersunk on the top.

Guard-bars of the form shown are to be securely bolted to the frames and buffer-plate.

Dimensions.

Thickness of frames (finished)	0 ft.	1 in.
Depth over leading bogie wheel	1 "	3 "
" between cylinders and driving-horns	1 "	8 "
" between driving and trailing wheels (open)	1 "	10¾ "
Greatest depth of plates	2 "	11¾ "
Distance from centre of bogie to front end of frame	4 "	10 "
Distance from centre of bogie to centre of driving-axle	9 "	10 "
Distance from centre of driving-axle to centre of trailing axle	8 "	4 "
Distance from centre of trailing-axle to hind end of frame	4 "	0 "
Extreme length of plates	27 "	0 "
Distance from centre of driving-axle to front of fire-box casing	1 "	10⅛ "
Distance between frames at leading end	3 "	9 "
" " " from cylinders to trailing end	4 "	0 "
Height of top of frame from rail	4 "	1½ "
Depth of buffer-plate	1 "	3 "
Length " "	7 "	6 "
Thickness " "	0 "	1¼ "
" of foot-plate	0 "	$\frac{5}{16}$ "
Extreme width of foot-plate	7 "	10 "

Outside frames to be of BB Staffordshire angle-iron (the step-plates to be welded on), and to be stayed to the inside frames as shown on drawings. All the rivets to be countersunk outside.

Section of angle-iron for frames 4 in. by 2½ in. by ½ in.

Cylinders to be made of the best close-grained hard and strong cold-blast cast-iron, twice cast, as hard as can be worked,

and perfectly free from honeycomb or other defects. They must be bored out perfectly true, the ends being bell-mouthed. The cylinders are to be made with loose covers at each end, the back cover having provision for carrying the front ends of slide-bars. All joints and faces to be machined and scraped to a true surface, so that a perfect joint can be obtained. When the cylinders are bolted together they must be tested by hydraulic pressure to 250 lbs. per square inch. The cylinders to be horizontal, and to be attached to the frames by flanges (the holes in which and in the frames are to be rosebitted), and secured by turned bolts a driving fit. The front flanges and covers are to project through the frames as shown on drawings. To be provided with waste-water cocks and gear worked from the left-hand side of foot-plate.

The top of cylinders to be covered with thin fire-brick or cement; the bottom flanges to be planed perfectly true, so that the bogie-pin plate may bear truly against them.

Dimensions.

Diameter	1 ft.	$5\frac{1}{2}$ in.
Stroke	2 "	2 "
Distance of centres	2 "	4 "
" " valve-spindle centres	0 "	$3\frac{1}{2}$ "
Thickness of metal	0 "	$\frac{7}{8}$ "
Length of ports	1 "	2 "
Width of steam-ports	0 "	$1\frac{1}{2}$ "
" " exhaust-ports	0 "	$3\frac{1}{2}$ "
Thickness of bridges	0 "	1 "
Length of working face	0 "	11 "
Distance from centre of driving-axle to centre of exhaust-port	9 "	10 "

Pistons to be of tough cast-iron, made from cylinder metal, and to be sound and free from all defects. To be accurately fitted to cones on ends of piston-rods, and fixed with nuts as shown on drawings. Piston-heads to be turned $\frac{1}{32}$ in. smaller than bore of cylinder. Packing-rings to be of cast-iron, turned

only on the outside and on edges, and made $\frac{1}{2}$ in. larger in diameter than cylinder bore, and then cut and sprung into their places. When finished the whole must be an easy but accurate fit in the cylinder, so that the piston and rod can be moved backwards and forwards by hand.

Dimensions.

Width of piston....................................	$3\frac{1}{4}$ in.
" " rings (two in each piston).................	$\frac{1}{2}$ "
Thickness of rings..................................	$\frac{1}{3}$ "

Piston-rods and cross-heads to be of the best mild cast-steel, manufactured by Vickers, Sons & Co., Cammell & Co., or other approved makers, with cone and nut for fixing to piston; the cross-head to be solid with the rod.

Dimensions.

Diameter of rod................................	0 ft.	$2\frac{3}{4}$ in.
Length between cone and shoulder of cross-head	3 "	$\frac{1}{8}$ "
Taper of cone in piston..................1 in 3	..	..
Number of threads per inch piston end.......6	..	..

Gudgeon pins to be of best Yorkshire iron, keyed in the cross-heads and well case-hardened.

Slide-bars and slide-blocks.—Slide-bars to be of cast-steel from the same makers as piston-rods, and to be provided with brass oil-siphons to drawings. The slide-blocks to be of cylinder metal, sound and free from all defects.

Dimensions.

Width of slide-bars..........................	0 ft.	3 in.
Thickness "	0 "	$2\frac{1}{4}$ "
Length of "	3 "	$8\frac{3}{4}$ "
" " slide-block..........................	1 "	0 "
Distance between slide-bars vertically..........	0 "	$3\frac{3}{8}$ "
" " " horizontally.........	0 "	$6\frac{7}{8}$ "

Connecting-rods to be of best Yorkshire iron, forged solid in one length. The brasses to be of gun-metal, those for the

big ends to be lined with white metal. The cotters to be of steel, and the bolts of the best Lowmoor iron forged from the solid; the heads must on no account be welded on.

Dimensions.

Distance of centres...........................	6 ft.	2 in.
Diameter of big-end bearings....................	0 "	$7\frac{3}{4}$ "
" " small-end bearings...............	0 "	3 "

Slide-valves and valve-spindles.—The valves to be of phosphor bronze by the Phosphor Bronze Company. The spindle frames and intermediate spindles to be of best Yorkshire iron, of the form shown on drawings, the latter to be well case-hardened.

The intermediate spindle-guides to be of cast-iron, bushed with white metal, and to have oil-boxes cast on as shown.

Dimensions.

Lap of valve....................................	1 in.
Lead (in full gear)..............................	$\frac{3}{32}$ "
Centre line of valve above centre-line of cylinder.....	1 "
Diameter of valve-spindle........................	$1\frac{7}{8}$ "
" " intermediate spindle..................	$3\frac{1}{2}$ "
Length of " " guides............	10 "

Valve-motion.—The valve-motion to be made from the best scrap-iron, and the working and rubbing surfaces to be thoroughly case-hardened, and provided with oil-siphons and grooves, and finished in the best manner. Expansion-link to be supported at the top from the forward eccentric-rod pin, the reversing-shaft being below the motion and behind the link. The motion-pins to be of best iron, thoroughly case-hardened and accurately fitted. Eccentric sheaves to be in two pieces, the smaller piece being of best scrap iron, and the larger piece of cylinder metal. Eccentric straps to be of wrought-iron, solid with the rod, and to be fitted with white-metal liners.

The following are specifications for a pumping-engine designed for the water-works of the city of Chicago, as called for about 1887:

SPECIFICATIONS.

For the construction, delivery, and erection upon and within the foundations, pump wells, etc., to be provided at the North Pumping Station of the Water Works of the city of Chicago, two (2) horizontal compound condensing engines, with the necessary appurtenances and equipment to render the whole in all respects fully complete, to satisfactorily perform the constant daily work in the manner and under the conditions hereinafter specified in connection with the city water supply.

The engines and appurtenances shall be exactly alike in form and proportions, and free as a whole, and in all their parts, from all that has not been proved by actual use and experience to be safe, reliable, and desirable.

The engines, including all their moving parts, together with the guard railing, exclusive of ample and convenient passage-ways around the outside and between the engines, shall not occupy a space greater than forty-five (45) feet in the line of motion, and fifty-nine (59) feet transversely.

Each engine, with not more than one-third ($\frac{1}{3}$) the boiler capacity to be furnished, shall have ample power and capacity to raise from the inlet chamber directly into the pipe-system of the city not less than twelve (12) million U.S. gallons of water in twenty-four (24) hours, under a head equivalent to one hundred and fifty (150) feet above the surface of the water in the said inlet chamber during the time of test.

The engines and all the appurtenances shall be so constructed that one engine may operate independent of the other; in such case they shall perform a due proportion of the work specified, in the manner and under the conditions above named. When both engines are working together, they shall deliver in like manner and under like conditions at least twenty-

four (24) million U. S. gallons each twenty-four (24) hours, for such continuous length of time, not exceeding six (6) months, for twenty-four (24) hours per day, as the Commissioner of Public Works may direct, and not to exceed two-thirds (⅔) of the boiler capacity shall be required to generate the steam necessary to deliver said twenty-four (24) million gallons.

The engines, whether working singly or together, shall develop a duty of not less than ninety (90) million pounds of water raised one (1) foot high, for each one hundred (100) pounds of coal consumed in the furnaces of the boilers for the generation of steam, for all purposes of operating the machinery as a whole and for such period of time, not exceeding six days, as the Commissioner of Public Works may direct.

In the performance of said duty and capacity the steam-pressure shall not exceed eighty (80) pounds per square inch in the boilers, and the piston speed of the engines shall not exceed two hundred (200) feet per minute.

No deductions of any kind whatsoever will be allowed in the estimate for "duty" or capacity.

The test of duty and capacity shall be made as often and at such times during the term of guarantee as the Commissioner of Public Works may direct, and the quantity of water pumped by the machinery shall be determined by Weir measurement or other such mode as may be satisfactory to the Commissioner of Public Works, and the test or tests of such capacity and duty shall be made under such regulations and by such persons as the said Commissioner may determine.

The boilers shall be horizontal, uniform in size, material, and construction, with ample water space, and heating and grate surface. The number of boilers and the mode of setting shall be such that each boiler may be used independently, or any number of them may be used in common. Provided, however, that the number of boilers and capacity of same shall be such that two-thirds (⅔) thereof shall be ample to generate steam with ordinary anthracite fuel and natural draught of chimney, to supply both engines and their appurtenances working together, when performing the aggregate maximum work speci-

fied, leaving one-third ($\frac{1}{3}$) the number of boilers out of use for cleaning and repairs.

The boilers shall be made of the best quality of steel or iron of proper proportion, thoroughly braced, and be constructed in a workmanlike and substantial manner, with ample facilities for cleaning and repairing.

The boilers shall be subject to the test prescribed by the rules of the United States Government, for boilers to carry a pressure of steam of eighty (80) pounds per square inch.

General Conditions.

All the material used in the engines, boilers, and their appurtenances shall be of the best quality of their several kinds, and the machinery as a whole and in detail shall be of such form, material and proportion as to ensure ample strength and wear incident to the respective parts, and all the labor shall be performed in a skilful and workmanlike manner.

The contractors shall furnish and put in place all necessary induction or suction and eduction or delivery pipes, extending from the inlet shaft to pumps, and from pumps to street-main adjacent to location of engines, and shall also furnish and put in place all necessary steam, exhaust, water, and other pipes of every kind necessary, and all steam-pipes shall be covered with approved material.

The sketch hereto annexed shows in outline the relative location of building, chimney, etc., and approximate line of delivery and suction pipes.

The work indicated in these specifications is intended to embrace the full and satisfactory construction and erection of the engines, boilers and appurtenances, of the style, capacity, and duty specified, complete in all their parts.

The contractors for said machinery will be held responsible for the adequacy and strength, and also for its satisfactory operation under the maximum work to be performed.

All the material, tools, and labor necessary to construct, transport, and erect upon the foundations in the city of Chicago, the said machinery, and to render the same complete,

shall be at the sole expense of the contractor, and be considered as merged in the contract price stipulated for said machinery.

All fees for any patented invention, article, or arrangement that may be used upon or in any manner connected with said machinery or boilers, shall also be included in the contract price.

The contractors shall guarantee the machinery, boilers, and appurtenances to perform the work specified without breakage, or other defect, for a period of one (1) year from and after the time they may be started for general use.

Bidders will furnish their own designs, and will submit, with their proposals, drawings showing the general plan of the engines, pumps, valves, and boilers with such clearness and fulness of detail as will permit a thorough comparison of the merits of the machinery proposed ; also specifications detailing the form, dimensions, and kind of material of the important parts of the machinery ; also the number of square feet of heating-surface in the proposed boilers, and the mode of setting them, and such other information as may be necessary.

Bidders will also specify the "duty" they will guarantee the engines to develop.

All the labor of construction and fitting must, as far as practicable, be done before being delivered on the premises of the pumping works, and the introduction of the machinery into the building, and erection there, must be so managed as not to interfere with or interrupt the regular and safe working, at all hours, of any of the existing pumping machinery.

All the stone and brick masonry constituting the foundations, also all the necessary stone-cutting, drilling, and fitting in connection therewith, will be done by the city of Chicago.

Provided, however, that the contractor will be required to furnish to the Commissioner of Public Works, drawings, showing plainly by figures, etc., the form and dimensions of the foundations, also the precise location and dimension of all openings for bolt-holes, pockets, and whatsoever may be necessary to correctly construct the said foundations and other

mason work, together with the dimensions of the smoke-chimney, size of flue, height of chimney, and mode of connecting boiler thereto.

The contractor shall also furnish, in due time, all the iron work or parts necessary to be built in said foundations, chimney, etc., during the progress of construction of same.

Estimates of the work in place will be made by the Commissioner of Public Works, from time to time, and vouchers for seventy-five (75) per cent of its value will be issued; the balance of twenty-five (25) per cent will be paid after a satisfactory test of the capacity and duty of the machinery shall have been made.

In case the engines should fail to develop satisfactorily the capacity and duty in the manner and under the conditions herein prescribed, or shall develop any inherent defect as to strength of parts or otherwise, the machinery entire will be subject to rejection, or to such reduction in the amount to be paid for same as the Commissioner of Public Works may determine.

The entire work herein contemplated shall be fully completed and ready for regular daily service on or before* .... 188., or as soon thereafter as the foundations are ready.

Proposals must be addressed to the Commissioner of Public Works, Chicago, Ill., indorsed "Proposals for Pumping Engines."

The Commissioner of Public Works reserves the right to reject any proposal not in accordance with the advertisement and the general specifications, or to reject all bids if he shall deem it for the best interest of the city of Chicago.

Bidders must present evidence satisfactory to the Commissioner of Public Works that he or they are fully competent, have the necessary facilities and pecuniary resources to perform the work required in a satisfactory manner, and within the time prescribed.

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* Bidders must fill this blank with the guaranteed time of completion.

Each proposal must be accompanied with a deposit of two thousand ($2000) dollars in currency, or a certified check on some Chicago bank in good standing, payable to the order of said Commissioner as security for the faithful execution of the contract within ten (10) days after it may be awarded, and the party or parties to whom the contract shall be awarded shall give bonds with satisfactory sureties to the Commissioner of Public Works for the faithful erection of the work in the manner and form prescribed by the contract and specifications.

PROPOSAL.

The undersigned propose to furnish all the labor and materials, of whatsoever kind, necessary for the construction and completion of proposed engines, boilers, and their appurtenances at the North Pumping Station of the Water Works of the city of Chicago, for the sum of........... dollars and ....cents, complete, according to the advertisement inviting proposals for the same, and according to plans and specifications herewith submitted.

In case the Commissioner of Public Works should award or tender the contract to the undersigned, in accordance with the above proposal, and the undersigned should fail or refuse to enter into a contract with approved sureties, to the satisfaction of said Commissioner, then the deposit of two thousand ($2000) dollars accompanying this proposal shall thereby be forfeited to the city of Chicago.

The undersigned hereby certify that ..he ..ha.. read the foregoing specifications, and that .... proposal for the work is based on the conditions and requirements embodied herein, and should the contract be awarded to ..h.., ..he.. agrees to execute the work in strict accordance therewith.

....
....
....
....

NOTE.—Companies or firms bidding for the preceding

described work must give the individual names of the persons comprising such company or firm, with their addresses.

222. The Performance—Power, Economy, Duty, and other results to be attained—are all often specified in the contract; and the verification of the guaranteed performance in all particulars constitutes, as a rule, the concluding part of the transaction. It is generally expected that the contractor or the builder will see the machinery in good condition, working smoothly and satisfactorily in all respects, and giving the full guaranteed power within the guaranteed cost in fuel and other materials consumed, and in money, before finally leaving it on the hands of the purchaser and user. The power must be ample to do the intended work without laboring or overstrain, easily and steadily; the economy should be all that could reasonably be asked under the circumstances; and the duty, where the economical performance is so gauged, should be fully up to the best practice of the time, if good work is demanded at all.

The following provisions are proposed for insertion in contracts for pumping-engines by a committee of the American Society of Mechanical Engineers, reporting a standard duty-trial scheme:*

In order that the contract between builder and purchaser of a pumping-engine may conform to the proposed standard, the guarantee as to performance should be expressed in the following terms:

(1) The engine shall perform a duty, based upon plunger-displacement, equivalent to not less than .... foot-pounds of work for each one million heat-units consumed.

(2) The leakage of the pump shall not exceed .... per cent of the total plunger-displacement when the engine is working at its rated capacity.

(3) The boiler shall supply one million heat-units to the engine on a consumption of .... pounds of .... coal, or it shall evaporate not less than .... pounds of water

* Transactions of American Society of Mechanical Engineers. 1889.

from and at 212 degrees per pound of the combustible portion of the coal named.

(4) The mode of determining these quantities is to conform to the standard method of conducting duty-trials recommended by the Committee of the American Society of Mechanical Engineers.

Should one contractor furnish the engine and another the boiler, separate guarantees will be made, the individual requirements of which are the same as those noted.

It is desirable, where both parties concur therein, to introduce into the contract the following additional provision regarding friction, viz.:

"The friction of the engine shall not exceed .... per cent of the indicated power developed in the steam-cylinders."

In power, it is generally demanded that a margin of not less than 20 or 25 per cent in excess may be given without sensible reduction of speed or loss of economy; in economy, the engine should usually equal that shown in the best practice, as, perhaps, not more than 30 pounds of steam or of ~~fuel~~-feed water, or about 3½ pounds fuel per horse-power and per hour for non-condensing engines of usual simple form and moderate sizes, two thirds this for good condensing engines, and not much above one half for the best compound types, all under such high pressures as are now familiar. A duty of 60 to 70 millions of foot-pounds per 100 pounds of coal for simple pumping-engines, 100 millions for compound, and 120 millions for triple-expansion, should be attainable under favorable conditions, and should be demanded in the specifications for the best cases.

The following is a good sample of specification for a non-condensing, high-pressure, compound, high-speed engine of small power, including a somewhat remarkable guarantee as to regulation, which guaranty, however, was certainly met:

SPECIFICATIONS FOR AUTOMATIC CUT-OFF TANDEM COMPOUND ENGINE. ....HORSE-POWER.

............189

We agree to furnish........horizontal automatic cut-off engine of the tandem compound type, from our latest patterns, especially adapted for.... ....purposes.

The rating of this engine is based on a cut-off of about $\frac{1}{8}$ in the high-pressure cylinder, and so arranged as to carry the expansion in the low-pressure cylinder to about atmospheric pressure.

A liberal reserve capacity above the rating is obtainable. The size of cylinders in each case will depend somewhat on the conditions in regard to high or low steam pressure, condensing or non-condensing.

Horse-power	....to....
Revolutions per minute	....to....
Diameter of high-pressure cylinder . . .	....inches
Diameter of low-pressure cylinder . . .	....inches
Length of stroke	....inches
Diameter of governor-wheel to be selected by you	and ....inches
Diameter of balance-wheel to be selected by you	and ....inches
Width of belt for governor-wheel . . .	....inches
Width of belt for balance-wheel . . .	....inches
Floor space, ...	....inches
Diameter of steam-pipe	....inches
Diameter of exhaust-pipe, ...	....inches
Weight	......lbs.

Before shipment of the engine we to furnish a template, together with foundation-bolts, nuts, and plate, and plan of foundation.

The list of fittings following are intended to include everything necessary to complete a first-class outfit in every respect:

One .... throttle-valve.

.... sight feed-oil cups with standards complete.

One eccentric oiler standard.
Two No. 0 brass oil-cups for intermediate.
Two No. 00 brass oil-cups for intermediate.
One .... cylinder-lubricator, nickel-plated.
Three milled wrenches, $\frac{5}{8}''$, $\frac{3}{4}''$, and 1″.
One wrench for piston.
One wrench for piston-nut.
One monkey-wrench.
One stuffing-box wrench.
One connecting-rod wrench.
One spanner for valve-stem gland.
Two wheel-keys.
One $\frac{3}{8}''$ nipple for drip-pipe for frame.
Four $\frac{1}{2}''$ nipples for cylinder drip.
Four $\frac{1}{2}''$ elbows.
Two $\frac{1}{2}''$ close nipples.
Four $\frac{1}{2}''$ globe valves.
One $\frac{3}{8}''$ cross T.
Six bolts and templet for foundation.
One balance-wheel.
One pulley containing governor.
One screwdriver for governor-springs.
One set of extra piston-rod packing.
One set of extra valve-stem packing
One gallon can of cylinder-oil.
One gallon can of engine-oil.
One engineer's card.
One book of instructions.
One $\frac{5}{8}''$ cap-lifter.
Two No. 2 brass oil-cups.

All wearing-parts of the engine having flat surfaces are scraped to surface plates, and the very finest material and workmanship is used in the construction of the engine.

In offering this engine we make the following guarantees:

1st. That the material and workmanship shall be the very best throughout.

2d. That its efficiency or fuel economy shall be the best obtainable for an engine of its type and horse-power capacity, and that it shall maintain this high grade of economy through a much longer period of time than is possible with any other form of this class of engine.

3d. That its regulation, as a result of the new principle, embodied in its construction, shall be better than has ever been realized by any other system of governing.

4th. That the engine shall not run one revolution slower when fully loaded than when running empty, and no reduction of boiler-pressure shall reduce the speed of the engine one revolution so long as there is sufficient boiler-pressure to move the load, the same result being obtained from either the governor or balance-wheel, or both.

223. **The Verification of the Contract,** as fulfilled by the contractor, becomes the first subject to be considered after the machinery has been completed, erected, and set in operation. It is then necessary to go over the contract in detail, comparing its requirements with the results attained and work done, and determining to what extent, if at all, the contractor has failed to complete his contract. This work of comparison is usually done by representatives of both parties acting in concert, and both the legal advisers and the consulting engineers may be called upon to settle the question. Such a revision of the contract and of the work of the contractor includes, often, both a careful inspection of the machinery and a formal trial. The inspection should verify every clause of the specifications relating to design and construction, and the trial should determine with precision and certainty whether the performance of the apparatus is all that was guaranteed, either as to power or to economy of operation.

Where disagreement occurs between the two parties to the contract, the matter in dispute is very commonly referred, by agreement of both sides, to referees, sometimes a single "expert," sometimes a board or commission consisting of three or more men of good professional standing, and trusted by both sides. In many cases of such reference, each side chooses its

representative, and the two select the third referee. Such referees are sometimes given only advising power, but they are oftener endowed with full authority to render a decision binding on both sides. In default of settlement in this manner the next recourse is to the courts.

224. **The Duty of the Inspector,** in greater detail, may be stated as follows :—

The first step is often taken during the period of construction, the inspector being appointed in advance, and entering upon his duties as soon as the purchase of materials and work upon them begin. He at once carefully studies the contract to familiarize himself with the stipulations made as to quality and tests of materials, and the method of construction and fitting, planning the best system possible of inspection and of test to meet the stated demands. He is present at the delivery of the material supplied the contractor, carefully inspects it visually and by test, rejecting the bad, setting aside the doubtful for further examination, and provisionally accepting and marking the good. The prescribed tests are then applied in the stated manner and all material successfully meeting the stipulated conditions is then formally accepted and so marked.

During the progress of the work of construction, the inspector measures the dimensions of each piece, in the rough and as finished, compares them with drawings and specification, and either passes or condemns them according to the result of this comparison. If in complete accordance with the drawings, and finished fully up to specification, the piece is accepted, and the contractor to that extent relieved from responsibility unless, as is often the case, the contract contains a clause holding him responsible for any defects discovered later, or within a specified time after delivery of the finished machine; in which case the piece is passed provisionally.

As the parts are "assembled," to make the completed machine, the inspector examines their fit, their details of connection, and, if practicable, their working, pair by pair. Any defect of form, size, finish, fit, or of operation so revealed leads to the rejection of the defective piece, and it must either be made

right, or a correctly made substitute furnished. Where the work has been made to gauge and on the interchangeable system, those pieces which passed inspection when gauged are very certain to be found right on assembling, and the inspector finds little to do at the latter stage.

The inspector finally examines the machine as a whole, as completed, and, all being found right, or, if wrong, made right, he makes his final report, and the purchaser accepts it, usually subject to the result of final trial. If the inspector be a man of experience, skill, good judgment, and thoroughly careful and conscientious, the final outcome of his labors will be an excellent piece of construction. But no builder of reputation will shield himself, under any circumstances, behind the report of an inspector, should it prove, later, that an error has been committed. He cannot afford to allow defective work to leave his hands, or any suggestion to stand as to his integrity. There rarely occurs an instance in which the builder is not glad to make good any defect traceable to his fault or that of his workmen, within any reasonable time after the delivery of the work. In exceptional instances, the inspector may require all his native strength of character to do his full duty by his employer; and this is one of the qualifications demanded of the applicant for such duty.

225. The Trial and Acceptance of the work follow the final report of the inspector on its construction. Trials are almost invariably made by an expert or a board of experts, and, on important contracts, they are made often very complete, and the reports are correspondingly elaborate. It is usual to fully specify the character and the method and extent of the trial in the original bargain and contract. The endeavor should be to make this test under conditions as nearly those of regular working as possible, and in such manner as to give exact and unquestionable measures of all quantities involved. Should several differing and independent sets of conditions be specified, it is sometimes necessary to make several similarly varying trials. In all cases the exact fulfilment of the contract is the main issue. The method of trial considered standard

and best in the case of important steam-engine contracts have been described elsewhere. They involve often the highest combination of scientific knowledge, experience in physical investigation and in mechanical work, and the operation of steam-engines and boilers.

In cases in which boilers are subject to the supervision of an inspection or an insurance company, the designs are also submitted for approval. In such instances the responsible officers of the company usually advise as to the size and number of boilers, and working pressure, and often send to take a plan of the site to ascertain how they can be best set and connected.* A specification is then drawn up, and tenders called for on printed forms. The work is kept under inspection, and the plates are tested, sometimes at the boilermaker's and sometimes at the platemaker's. When complete, the boiler is inspected internally and externally, fittings included, tested by hydraulic pressure, and gaugings taken to see if any permanent set occurs. When the boiler is set, the setting is examined before steam is raised. The maker should draw up a general plan to show how the boiler should be set, how the piping should be arranged, and how connections are to be made, subject to approval.

226. Subsequent Liabilities, the contract being fulfilled and the machinery formally accepted and paid for, depend in part on the terms of the contract itself. It may hold the contractor liable for any extraordinary costs of operation or repair, or even for ordinary repairs, for a specified time, and for damages resulting from the use of defective material or

* The Josiah Parkes (1820) "Split Bridge," in use with soft coal, may not save coal, but it need not cause waste. The firemen need not, as a rule, apprehend that the admission of fresh air at the split bridge will waste fuel. Trials show that the constant admission of air above the fire, through an aperture of one seventieth the area of the grate, does not reduce economy.

A hanging bridge three feet behind the ordinary bridge, carried down to the centre of the flue, will assist in mingling the gases and in the prevention of smoke. If the draught be good, the smoke may be so reduced as to be unobjectionable.

from bad workmanship; but, in any case, the principle that the common law is common sense is likely to settle fairly well the final limit. A contractor, unless expressly relieved by his contract, can never escape any serious consequences of evident neglect, incapacity, or dishonesty, and builders of integrity and of good repute never seek to evade responsibility for even inadvertent errors or defects of workmanship.

CHAPTER VIII.

FINANCE.

227. Financial Considerations come in in every case to affect the operations of the engineer in the application of his art. His problem is never to simply apply machinery to the transformation, the transmission, and the useful application of energy in the best possible manner, mechanically considered: it is always the devising of a method and of machinery and apparatus to transform and to make useful natural stores of energy in, on the whole and all conditions considered, the most economical and productive manner. In studying this problem the balance-sheet constructed should never exhibit simply the value of product and cost of production measured in expense of operation of the machinery itself, alone; but it properly includes every cost, direct and indirect, which enters the final result. The costs of a machine, and its operation, are not simply to be taken as the amount paid for it to its makers, and the expense of attendance and of lubrication: they include costs of transportation, of mounting, of housing, rent of space occupied, any increased insurance; the running expense must be charged, not only with costs of attendance, but also with interest on cost, and on all expenditures involved in its purchase, installation, and use; with deterioration, repairs and renewals, and every item that enters the bookkeeper's accounts that appears there as a consequence of the introduction of this machinery. The question is one which the proprietor finds of vital importance, and is really, How is the total balance-sheet of the business affected on either side by the introduction and employment of this machinery?

In selecting engines with reference to costs it will be found that their prices per horse-power usually vary with size on either side a minimum, which minimum with makers of large engines is at 300 to 400 horse-power. Hence, a pair of engines

may even be found to be less costly than a single engine of equal power. Where this is the fact the attendant advantages of reduced size and weight of fly-wheel may conspire to make their use desirable or imperative. In compounding, this consideration has added force.

The finance of the case often determines the choice between widely different plans and alternate schemes in engineering practice; as when, according to Mr. Fitzgerald, in choosing the system of water-supply for a town a gravity system would be selected whenever quantity and purity may be relied upon for the average life of the "plant," and when about 8 per cent on cost will not exceed about 60 per cent of the operating expense. He would select a direct-pumping system when the town exceeds in demand a half-million gallons, the supply is good and ample, but the country level, so that an elevated reservoir would not be practicable; while for smaller places a direct system, with a very large stand-pipe to serve as a reservoir, is advised.*

The use of the steam-jacket, even, may be found advisable or not, accordingly as fuel and construction are of great or of little relative cost, or as saturated or superheated steam is employed, and as the temperature of the interior surfaces of the engine is thus made more or less a function of jacket-action.†

In many cases no hesitation is felt by the engineer, and no computation is needed to decide the question of adopting proposed changes, and at a known cost: as, for example, as has actually occurred, where a locomotive doing heavy work can, by the reduction of the diameter of its stack ten or fifteen per cent, with increase of its exhaust-nozzle of similar amount, be made to exhibit at this small cost a gain of 25 per cent in coal

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* Trans. Am. Soc. C. E., April 1891.

† Small variations of temperature may produce large economies. Mr. Donkin found a gain of 26 per cent due 49° F. rise of temperature produced by jacketing and with superheated steam, and a gain of 31 per cent with saturated steam, the increase of temperature being 34° F.; the net gain varying in a number of such cases from 23 to 40 per cent with a condensing engine. Non-condensing, the gain was 43 per cent, with a difference of temperature of 47° F., using saturated steam.

See Proc. Brit. Inst. C. E., vol. CVI., p. 275. See also this work, Part I, Chap. VI, §§ 152–166.

and steam consumption, by a drop in back-pressure of about one atmosphere.*

In other cases the intricacy of the problem and the magnitude of the costs involved may compel the most careful and minute study; as, for example, in the instances illustrated in the last chapter of Part I of this work.

Costs come in on all sides to determine or to modify the final plans of the engineer. The choice of location; the plans of the establishment and of all its machinery; questions as to advisability of substituting new for old machines; whether a compound shall replace a simple engine; whether a new engine shall be obtained or an old one shall be compounded,—all these, and many more are raised and settled by financial considerations. The Author has, when engaged in construction, sometimes found it an advantageous plan to substitute a new cylinder for an old one, converting an engine of older and less economical form into a more modern "drop cut-off" or other more useful and efficient machine, with comparatively small cost in alteration and very large ultimate financial gain.

228. Finance, in Steam-engineering, thus becomes an important matter to the designer and to the constructing engineer, as well as to the user of the machine. When it is known that fuel is cheap, for example, it becomes evident that it will not be wise to go to any considerable expense to obtain a very large boiler, or an engine having complicated valve-gear or other costly parts especially intended to give great economy. On the other hand, if it is found that the fuel is expensive at the point at which the machinery is to be located, it may be advisable to go to very great expense in these directions to insure its economical operation.

Again, if the costs of installation are great, or if rents are high; if repairs cannot be made easily and cheaply, or if no skilled workmen or reliable attendants are to be found at the stated locality,—it becomes a wise plan to provide the simplest and strongest type of engine, even at the expense of efficiency.

In any event, the aspect of the case to be considered is not

* In this instance the stack was given the form of the contracted vein, the diameter stated being that of the throat.

the first cost, the outgo at starting a steam-plant, but the relation of the total of all outgoes, during the life of the machinery or during a sufficiently extended series of years, to the total of all receipts in the same time. If such a comparison indicates a good business profit, the operation is a satisfactory one; otherwise, it is an unfortunate one. If it can be shown that the plan adopted and the system of machinery employed gives the best possible return, maximum financial efficiency, as compared with any possible or proposed modifications of the plan and of proportions of the machinery, the conclusion follows that the scheme adopted is most creditable to the designer and most perfectly in accord with the proprietor's interests, whatever the apparent merit or the actual efficiency of any part of it.

229. **Costs and Economies,** receipts and expenditures, must thus be very carefully analyzed before it can be ascertained whether the designer, the builder, and the buyer have attained a creditable result. The real test is, in all cases, the final influence of the machinery employed on the profits of the business, and it may thus be necessary to study many matters quite remote from the apparatus itself and its own efficiencies. The progress of the marine steam-engine, since Watt, has been determined, not simply by the changes noted in its continual gain of efficiency, but also and largely by the value of stowage space on board ship, by the freight charges, and by the cost of construction and the power of propulsion of the ship itself, as modified by the progress of all industries during this interval. The question asked of the engineer is now to make the cost per ton per mile, or per passenger per mile, a minimum on the railway or on ship board. Large ocean steamships have during the ten years just passed cost vastly less, on this basis, than in the preceding decade, and the average gain during the last period is probably not less than 20 per cent; the cost of British "liners," per ship and per mile, having fallen from about \$5.00 to \$4.00, as minima, on all accounts, including regular charges, insurance, repairs and depreciation, while speeds have been at the same time increased in an equal ratio. The gross expenditure incurred, as above, in transpor-

tation being compared with the receipts at current rates of charge, the difference is a profit which determines the magnitudes of dividends paid. On this net return, the costs, not of fuel or steam, but of the machinery on all accounts, have a very important influence. It is therefore necessary, in all cases, to analyze these costs very carefully, completely, and accurately.

It is important to arrange a good system of record and report, where costs are to be closely determined. Mr. Towne has published a set of record-blanks and of data thus obtained which accomplish the desired end satisfactorily, as follows :*

ENGINE-ROOM REPORT.

For....................189..

BOILERS.

Date.	Fuel.						Number of Boilers in Use. 5.				Amount of Feed-water, Gallons.	Remarks as to Fuel.
	Bituminous Coal, Pounds.	Number of Lot.	Anthracite Coal, Pounds.	Number of Lot.	Cinders and Coke, Pounds.	Hours Burning Wood, etc. One Boiler.	Day. Banked.	Day. Fired.	Night. Banked.	Night. Fired.		
							Hours for One Boiler.					
1	......	...	.....	.	....	...	....	33	9	30	....	No. — was a bad
28	......	..	.....	..	....	....	25	25	40	30	...	lot of anthracite
3	20,700	3	20,700	21	....	....	....	55	30	35	....	dust, and necessi-
4	10,800	3	10,800	21	....	. .	....	55	30	35	...	tated using equal
5	8,700	3	7,800	21	1,200	....	...	55	35	30	....	parts of the two
												kinds of coal.

The above is the heading of the report. The columns are all footed up properly, when the blank is filled. Other records are as follow :

USE OF STEAM.

Date.	Main Engine running Hours.	Pump. Number of Strokes.	Heating Buildings.							Thermometer.			Wind Average.		Remarks: as to Weather, etc.
			Exhaust		Back-pressure. Pounds.	Live Steam.									
						Day.		Night.							
			Hours.	Opening.		Hours.	Opening.	Hours.	Opening.	7 A.M.	12 M.	6 P.M.	Direction.	Force	
1	..	....	...	..	...	....	...	..	..	..	....	...	....	...	
28	....	....	...	..	...	.	...	10	½	14	24	12	....	...	
3	10	4,366	10	½	3	10	½	8	½	10	20	14	N.W.	S	
4	10	3,990	10	½	3	10	½	7	½	10	20	24	E.	L	
5	10	5,228	10	½	3	10	½	6	½	18	40	36	E.	S	

*Trans. Am. Soc. M. E., vol. VIII, No. CCLV.

INDICATED HORSE-POWER.

(To be taken every Saturday.)

Date.	9 A.M.	10 A.M.	11 A.M.	12.03 M.	2 P.M.	3 P.M.	4 P.M.	4.50 P.M.	Remarks.
8	139.7	142.5	141.8	......	134.87	136.50	134.00	71.5	
15	146.2	143.6	142.18	......	130.00	121.87	126.25	66.62	
22	141.37	141.37	141.37	......	125.98	125.12	136.62	67.37	
29	133.24	138.10	131.62	......	121.87	130.00	135.12	61.75	
Totals.. ..	560 51	565.57	556.97	......	512.72	513.49	531.99	267.24	
Means......	140.13	141.39	139.24	... ..	128.18	128 37	132.99	66.81	

MAIN ENGINE.

CYLINDER, 20 INCHES; STROKE, 42 INCHES.

Average revolutions per minute 72

Average pressure of steam in boilers, in lbs............... 62

Average horse-power, as above, during working hours.. 135.05

Average horse-power, as above, after working hours 66.81

WATER.

Total number of strokes of pumps..... .... 100.945

Total number of gallons pumped, at 3 gallons per stroke.... 576,735

Assumed cost per 1,000 gallons.................. .... .05

Total cost for month.................... .... 28 84

.............., 189... ...., Engineer.

Remarks... ....

SUMMARY OF CONSUMPTION OF FUEL.

Kind of Fuel.	Pounds.	Proportion. Per cent.	Tons of 2,240 Pounds.	Average Price per Ton.	Cost.	Remarks.
Bituminous coal..	145,200	34 39	64.82	$3 90	$252 80	
Anthracite coal.............	272,700	64.58	121.74	2 37	288 85	
Cinders, coke, etc...........	3,000	.71	1.34	60	80	
Wood, shavings, etc..... ...	....	*.32	.. .	......	†1 76	
Totals..	420,900	100.00	187.90	Mean, $2 89	$544 21	

* To ascertain proportion of wood, find (1) total number of hours *all* the boilers are fired, and (2) total number of hours burning wood; then divide (2) by (1), and result will be proportion desired.

† To ascertain value of wood burned, divide total number of hours run of boilers burning coal and cinders into cost of latter. Then assume *value per hour* thus ascertained as the value of the wood fuel, and multiply this by the number of hours burning wood.

STEAM.

Average weight of fuel consumed per square foot of grate, per hour		9.10 lbs.			
Average evaporation of water per square foot of grate, per hour		7.25 gals.			
Average evaporation of water per lb. of fuel burned		.799 gals. = 6.66 lbs.			
Total gallons of water evaporated		336,510.			
Main engine, number of hours run	246				
Average horse-power	135.05				
Aggregate horse-power for month	3,376.25				
Gallons water per horse-power per hour	3.60 (= 30 lbs.)				
Total gallons chargeable to engine		119,600			
Well-pumps, number gallons pumped	576,735				
Assumed equivalent	.00393				
Total gallons chargeable to pumps		2,266			
Heating, balance of water evaporated:					
Total gallons chargeable to heating		214,664			
TOTAL GALLONS		336,510			
Percentage of steam chargeable to Power				35.54	$238 25
" " " " " Pumping				.68	4 55
" " " " " Heating				63.78	427 55
				100.	$670 35

Heating, cost of:

Steam, as above	$427 55
Wages	47 31
	$474 85

SUMMARY.

Boilers.		Power.	
Cost of fuel for month	$544 21	Steam, 35.54 per cent of total cost	$238 25
Wages of firemen	119 06	Wages	31 11
Wages of night firemen		Oil, 9 gallons cylinder-oil at 80 cents	7 20
Water, 15 per ct. of evaporation, say 50,476 gals. at 0.5 per 1,000	2 52	Oil, 2 gallons machinery-oil at 40 cents	80
Gas, 32 per cent of total 1,600 cubic feet at $1.80	2 88	Waste, 10 lbs. at 9½ cents	95
Repairs during month	1 68	Gas, 32 per cent of total 1,600 cubic feet at $1 80	2 88
Sundry supplies		Repairs during month	
		Sundry charges	
Total cost of steam	$670 35	Total cost of power	$281 19

Mr. Burleigh proposes the following blank for keeping accounts of this class: *

CAMDEN LIGHTING AND HEATING CO.

Statement of Expenses for................189..

Heads of Accounts.	Arc Lighting.	Inc. Lighting.	Power.		Total.
			Street Cars.	Stationary.	
Boilers, repairs of..................................					
Belting, " "					
Boiler-house and stack, repairs of..					
Carbons					
Converters, repairs of.......................... ...					
Clerks					
Dynamo, repairs of					
Dynamo attendants..					
Enginemen and firemen					
Engine, repairs of................................					
Fuel					
General officers' salaries....					
Horses, wagons, and harness..........					
Insurance.					
Interest on notes, bonds, and mortgages..........					
Incidentals					
Instruments of all kinds................					
Lamps, repairs of					
Lamp supports and fixtures......................					
Lamp-globes.....					
Lamps, incandescent....					
Linemen					
Loss and damage					
Labor at station............................					
Labor on street-cars					
Legal expenses....					
Meters, repairs of..................................					
Motors, " "					
Oil..........................					
Office expenses, repairs and furniture.............					
Poles and lines, repairs and renewals..					
Right of way					
Station, repairs of and furniture for...					
Stationery and printing					
Superintendent and foremen					
Steam-piping					
Shafting...........					
Taxes, city					
Taxes, State					
Tools, repairs and renewals.........					
Trimmers and inspectors..........................					
Water............					
Waste					
Total expenses for month..					

Total average arc-lights............ Total inc. output in ampere hours.... ...
Total arc-light hours................ Total as shown by meters ....
Total cost per arc-light hour........ Total loss..........................
Total cost per 100 ampere hours..........
Total stationary output in watt hours.........
Total as shown by meters.................. ...
Total loss.... ... .
Total cost per 1000 watt hours.................

REMARKS:

....
....

* Trans. Nat. El. Lt. Assoc.; Montreal Meeting, 1891.

The following is a system of record for a large electric-lighting station, as arranged by Mr. Decamp:*

Revenue.	Average Cost.	Average Revenue.
Lights		
Other sources		
Total revenue		
Less rebates, etc		
Expenses.		
Pay-roll, electrical department		
" motive-power department		
" office		
General stationery		
" cartage		
" expressage		
Horses		
Tools		
Miscellaneous		
Taxes, licenses, etc		
Interest		
Extraordinary		
Total general expenses		
Coal		
Carbons		
Engines, oil (machinery)		
" " (cylinder)		
" waste		
" belts		
" shafting		
" miscellaneous		
Boilers, furnaces		
" tubes		
" piping		
" pumps		
" miscellaneous		
Dynamo, oil		
" waste		
" belts		
" brushes		
" armatures		
" commutators		
" segments		
" miscellaneous		
Lamps, spools		
" cut-outs		
" armatures		
" carbon-holders		
" carbon-rods		
" insulators		
" globes		
" miscellaneous		
" incandescent		
Line, poles		
" wire		
" miscellaneous		
Net results		
Previously reported		
Total for the year		
Daily average (Sundays excepted)		
Sunday "		
Total hours burned......Average		

* Trans. National Electric Light Association, 1890; "Costs of Product at Central Stations."

Mr. T. C. Smith adopted the following:*

All work, of whatever kind done, should be charged to some account in the ledger, and a convenient division of accounts to which items should be charged will be found to be as follows:

Construction.—All poles, wire, insulators, cross-arms, and other material used in the construction of lines, whether overhead or underground. All fixtures, cranes, sockets, or wiring for incandescent lamps; and in general everything attached to a customer's premises, including any tools used for this work.

Machinery and Fixtures.—All dynamos and parts of same; belting, station instruments, and switches; and all other movable apparatus in the station.

Motors.—Where the company owns such.

Office Fixtures and Furniture.—Including all furniture in and about the station.

Station.—All real estate, buildings, engines, boilers, foundations, pumps, piping, shafting, and any fixed scales or machinery.

Patent and Legal Expenses.—Of every nature and kind.

Store-room.—All material and supplies of whatever kind to be charged to the several departments as issued. These various items should be, of course, considered as unavailable assets.

In what are known generally as the "Expense Accounts," a convenient division will be found as follows:

Advertising.

Carbons.

Fuel.

Insurance.

Interest and Discount (if any).

Interest on Bonds or Mortgages.

Lighting Supplies.—Including incandescent lamps for renewals, shades, globes for arc-lamps and dynamo-brushes, and other material of this nature.

* Ibid.

Machinery Repairs.—All repairs to dynamos, armatures, station instruments or switches, arc-lamps or parts of same.

Maintenance Repairs.—Being repairs to poles, lines, etc., converters in service, meters, and other appliances used or located outside the station.

Shop and Station Repairs.—Repairs to buildings, boilers and engines, piping, or any fixed machinery.

Printing and Stationery.—Of every kind and nature.

Salaries.—To the officers of the company.

Wages.—Of the employés, to be divided proportionately to each department, as explained above by the system of order numbers.

Oil and Waste.—Of every kind.

Taxes.—Of every kind.

230. **Costs of Engine and Expenses of Operation** may easily be sought out and evaluated in any actual case; but no general statement with specific figures can be given. The following examples will suffice to show what they are and what they amount to in good ordinary practice. In locomotive practice, as an illustration, it is customary, on many railroads, to prepare monthly statements of such expenses as can be conveniently obtained and grouped, in order to ascertain who is to be credited with most skilful management and what are the costs to be charged against each engine and its attendants, and the known cost of original construction or purchase affords a basis for the interest account. In some cases very complete analyses have been made of the whole cost account, as by Mr. Emery and others, in cases elsewhere mentioned.

The work of the steam-engine in the operation of electric railways can be compared only to that of the rolling-mill. The attempt is more promising of success when large generators in units of 300 to 500 horse-power can be used. What is wanted in the generating station for electricity is the smallest division of units consistent with the safe and economical operation of the plant. "High-speed automatic engines" can be successfully operated if they have large bearings and fly-wheel

capacity. On a cross-compound engine of, say, 300 horse-power, according to Mr. C. J. Field, there should be 6 to 8 tons in the fly-wheel, the bearings not less than 7 or 8 inches in diameter and 15 or 18 inches in length. A type of engine, in units of 500 horse-power, built at a rotative speed of about 140 or 150 revolutions, and with a piston-speed of about 650 to 700 feet per minute, is considered desirable, but is thought to be too fast for the detachable valve and too slow for the shaft-governor. In large work, economy of steam becomes the controlling consideration.

Direct connection to dynamos is always advised where possible. Even a "drop cut-off" engine of 300 to 500 H.P. at 100 to 80 revolutions per minute may be so connected in many cases. On such engines of 500 H.P., a fly-wheel of approximately 60,000 lbs. weight is needed. On engines working at 150 revolutions, 30,000 to 40,000 lbs.

Engines should be suitably proportioned to their work, to avoid excessive condensation and re-evaporation. Economy often dictates the substitution of a small for a large engine, where the latter is too lightly loaded, or, in some cases, the cylinders should be bushed to reduce their size and condensing effect. In one such case the owner stated that "the saving effected had amounted to 50 per cent in coal and water, and that the saving of water alone paid for the alterations." *

Introducing, in another case, a compound engine in place of a single-cylinder, both condensing, the owner reported that the cost of the engine, engine-house, and necessary alterations was approximately $5500 (£1100). Previous to the alteration the yearly coal consumption was 1373 tons. With the compound engine the yearly consumption was but 530 tons. The saving of coal during eleven years was 9269 tons of best steam-

* In this case the engine was non-condensing, and had a diameter of 15 inches, a stroke of 2 ft. 6 in., a speed of 81 revolutions, and the boiler-pressure was 60 lbs. The diameter of the cylinder was reduced to 12 in., giving 64 per cent of the original area, the stroke, the revolutions, and the pressure remaining the same. (L. E. Fletcher, 1891.)

coal, which amounts to $1895 (£379) yearly, driving an increased load.*

In another instance, a pair of tandem double-cylinder compound engines was made by adding a third cylinder to each, and raising the pressure from 60 to 150 lbs.

The dimensions of the cylinders were:

	Diameter.	Stroke.	Revolutions.	Relative Volume.
High-pressure.....	15 in.	6 ft.	36	1.00
Intermediate......	23 in.	6 ft.	36	2.35
Low-pressure......	40 in.	6 ft.	36	7.11

The gross coal-consumption fell from 60 tons a week to 45, or from 2.77 lbs. per I. H. P. to 2, and the saving amounted to 25 per cent. Experience indicates that in condensing engines it is wise to adopt the triple-expansion system if the pressure exceeds 120 lbs. on the inch, and in non-condensing engines 150 lbs., with the double-cylinder.

According to Mr. Chas. J. Field, the costs of power and installation of systems of transmissions were, for the best practice in New York, 1890–91, substantially as follows:

Type.	Pounds of Coal per H. P. hour.	Cost per H. P. Sizes over 100 H. P.	This is based on an evaporation of 9 lbs. of water per lb. of coal.
High-speed, single................	4 to 5	$11 to 13	
" " compound...........	3 " 3½	14 " 16	
" " " condensing.	2¼ " 2½		
" " triple...............	1¾ " 2	18 " 22	
Corliss, single....................	3½ " 4	16 " 18	
" compound...............			
" condensing.............	1⅔ " 2	22 " 25	
" triple....................	1½ " 1⅔	27 " 30	

The cost of steam-plant complete is for high-speed $45.00 to $55.00 per H. P., and for Corliss, $65.00 to $75.00 per H. P.

The number of units of power in a power-plant should be as few as is consistent with safety and economy of operation, but not less than two.

* Ibid.

The relative commercial economy of engines and cost are as follows:

There are three classes of boilers.

(1) Horizontal return-tubular, which is the most general in use, and costs $9.00 to $10.00 per H. P.

(2) Vertical tubular boiler with water-leg, giving an internal fire-box, economical in floor-space, largely used throughout New England. Cost $10.00 to $12.00 per H. P.

(3) Sectional or water-tube boiler, especially adapted for higher pressures and safety. Cost $17.00 to $19.00 per H. P.

Builders often issue circulars giving dimensions and prices which serve to exhibit relative values; but they are usually subject to a discount, the amount of which varies with the state of trade, and must always be investigated by the engineer seeking to solve such problems. The opposite page supplies an illustration obtained from an English firm building small marine engines in 1892.

The size of crew needed to handle steam-machinery is dependent upon the size and power, and also on the efficiency of the machinery. With very small powers, as up to 100 or 200 horse-power, one man may act as both engineer and fireman, and the cost of attendance is then usually a constant quantity. For larger powers, it will require one man in the boiler-room, at least, for each ton of coal handled per hour, one engineer to take charge of the whole and to handle the engine, and, with very large engines, one or more water-tenders in the boiler-room and oilers in the engine-room. The accompanying table, kindly prepared for the Author by Mr. H. P. Norton, U. S. N., shows the number of attendants on several large ships. As is here seen, one attendant is needed for each 200 to 250 square feet of heating surface, or to 10 or 15 feet of grate-surface, although the proportion varies greatly. The number of engineer-officers has no direct relation to the size of the crew, but is determined by the number and arrangement of the engines.

In transatlantic steamers of high power (20,000 I. H. P.) and speed, it is found advisable to have a chief-engineer, 18 assistant-engineers for main engines, 7 assistants for the auxiliary

COMPOUND SURFACE-CONDENSING ENGINES.

Engines—100 lbs. pressure:										
Indicated horse-power..........	20	40	50	70	100	125	175	200	300	500
Diam. of high-pressure cylinder..	5 in.	7 in.	8 in.	9 in.	11 in.	12 in.	14 in.	15 in.	18 in.	23½ in.
Diam. of low-pressure cylinder..	10 in.	14 in.	16 in.	18 in.	22 in.	24 in.	28 in.	30 in.	36 in.	47 in.
Length of stroke...............	6 in.	8 in.	9 in.	14 in.	15 in.	18 in.	18 in.	18 in.	24 in.	27 in.
Shafting—centre of low-pressure cylinder to centre of propeller, in feet.	6	7	9	12	14	16	20	20	24	28
Price of engine only......	£178	£205	£240	£305	£430	£493	£670	£765	£925	£1680
Price of jet-condensing engine...	£164	£177	£205	£255	£360	£410	£555	£632	£735	£1350
Accessories:										
Price of propeller, stern-tube, and shafting, etc.	£18	£22	£28	£37	£53	£70	£95	£120	£155	£210
Price of do. with gun-metal propeller and stern-tube.	£30	£39	£49	£65	£96	£130	£176	£200	£300	£400
Price of copper piping to form connections for machinery.	£10	£13	£16	£22	£25	£32	£42	£52	£100	£200
Price of donkey-pump and cocks.	£16	£16	£21	£26	£28	£31	£31	£31	£32	£37
Price of sea-valves...	£12	£14	£15	£16	£18	£21	£31	£41	£42	£58
Price of outfit.....	£10	£11	£11	£13	£15	£16	£21	£26	£35	£47
Approximate weight, engine only	30 cwt.	40 cwt.	50 cwt.	90 cwt.	132 cwt.	160 cwt.	210 cwt.	240 cwt.	360 cwt.	600 cwt.
Approx. weight of accessories..	20 cwt.	25 cwt.	30 cwt.	35 cwt.	37 cwt.	40 cwt.	50 cwt.	52 cwt.	100 cwt.	260 cwt.
Boilers—100 lbs. pressure:									2 boilers each	2 boilers each
Diameter	3 ft. 9 in.	5 ft. 0 in.	5 ft. 4 in.	6 ft. 6 in.	7 ft. 0 in.	7 ft. 0 in.	8 ft. 0 in.	9 ft. 0 in.	8 ft. 0 in.	10 ft. 0 in.
Length................	5 ft. 0 in.	6 ft. 0 in.	6 ft. 6 in.	7 ft. 6 in.	8 ft. 0 in.	9 ft. 0 in.	9 ft. 0 in.	8 ft. 6 in.	8 ft. 0 in.	9 ft. 0 in.
Number of furnaces	1	1	1	1	1	1	2	2	2 each	2 each
Heating-surface in square feet...	90	160	193	250	425	470	566	670	1040	1780
Price with mountings, funnel, and uptake.	£90	£135	£147	£240	£320	£345	£390	£460	£735	£1235
Approximate weight, boiler only.	30 cwt.	55 cwt.	60 cwt.	82 cwt.	125 cwt.	140 cwt.	168 cwt.	210 cwt.	310 cwt.	520 cwt.
Approximate weight, boiler and mountings.	45 cwt.	75 cwt.	90 cwt.	120 cwt.	170 cwt.	185 cwt.	233 cwt	280 cwt.	430 cwt.	690 cwt.

machinery, 9 water-tenders, 36 oilers, 1 store-keeper, 54 firemen, 54 coal-heavers; total, 180. Another and smaller vessel carries 1 chief engineer, 14 assistant engineers for main engines, 2 assistants for auxiliaries, 12 cadets or apprentices, 9 water-tenders, 6 oilers, 54 firemen, 54 coal-heavers; total, 152.

Name of Vessel.	Engineer-officers.	Firemen.	Coal-passers.	Grate-surface, sq. ft.	Heating-surface, sq. ft.	Total
Eider (German)..........	8	54		756	19,698	62
Ems "	7	53		780	19,700	60
Buffalo (English).........	6	18		324	9,751	24
Bourgogne (French)......	8	79		900	23,000	87
Champagne "	7	90		830	23,000	97
		3 water	tenders			
Saale, Trave	13	24	24	799.5	22,630	68
		2 electri	cians			
Oregon.................	10	116		1,542	....	128
		2 electri	cians.			
America.................		92		917	22,926	94
American—						
		3 water	tenders,	messboys,	etc., 4	
City of Para.............	4	9	9	360	9,900	29
Acapulco, Colon	4	16		228	5,480	20
Newport................	4	25		426	10,650	29
Louisiana................	4	19		297	12,800	23

Another ship has 1 chief-engineer, 16 assistant-engineers for main engines, 6 assistants for auxiliaries, 2 store-keepers with 2 assistants, 24 oilers, 6 leading stokers, 60 firemen, 42 coal-heavers; total, 159. The single-screw ship of 6500 I. H. P. has 66 firemen and coal-heavers, and a total number of 89. "Blacksmiths and coppersmiths must also be added to the engineer force. The blacksmith is perhaps the most generally useful mechanic on board ship. He makes and repairs the tools, and it would be impossible to carry on work in the engine-department without the forge.

It needs but a glance at the great tangle of copper piping, miles in length, of all sizes and shapes, on all modern ships, to recognize the necessity of having a coppersmith in the engineer force.*

* Report of the Engineer-in-Chief of the Navy, 1890.

The following is the British naval allowance table for engine-room artificers on large ships:

Total No.	No. of each Trade.				Chief-stokers.
	Fitters.	Boiler-makers.	Engine-smiths.	Copper-smiths.	
2	1	1	....	....	....
4	3	1	....	....	....
6	3	2	1	....	1
8	4	2	1	1	1
10	5	3	1	1	2
12	7	3	1	1	2

About ten per cent of the "stokers" or "firemen" should be mechanics.

Mr. Powers makes the following comparison of the costs of operation, at then current prices, of various familiar types of steam-engine: *

STEAM-CONSUMPTION.

		Lbs. per hr. per I. H. P.
A	Single-valve, high-speed	28 to 32
B	Automatic expansion (Corliss), non-condensing....	24 " 26
C	" " condensing.......	20 " 21
D	" " " compound	15 " 16

COAL-CONSUMPTION; EVAP. 8 to 1.

	Lbs. per I. H. P. per hr.	Per cent.
A	3.50 to 4.00	100
B.........................	3.00 " 3.25	62
C.........................	2.50 " 2.62	56
D.........................	1.87 " 2.00	44

* Electrical Engineer; July, 1888. These figures cannot, of course, be accepted at other times and in presumably different states of the market.

FIRST COSTS PER I. H. P. (*Engines and Boilers*).

		Per cent.
A	$31 to $36	100
B	42 " 46	131
C	43 " 48	136
D	52 " 57	163

Boilers only.

A, $16 to $18; *B*, $12 to $14; *C*, $10 to $12; *D*, $7 to $9.

COST OF COAL AT $3.00 PER TON.

A cost per day, 400 I. H. P.	$24 75
B " " " " "	18 90
C " " " " "	15 24
D " " " " "	11 64

Comparing cost for a flour-mill making 100 barrels per day, employing 50 I. H. P., the following are the results:*

COSTS.

A	"Plant"	$1500	Int.	$90	Fuel per yr.	$2011
B	"	2700	"	162	" "	1638
C	"	3200	"	192	" "	1260
D	"	4300	"	258	" "	1008

TOTAL ANNUAL COSTS.

A, $2106; *B*, $1800; *C*, $1452; *D*, $1266.

These figures are modified by an indeterminate amount when the costs of machinery of transmission are considered and the effects of variations of total and partial loads are introduced,—sometimes, as in electric-lighting, probably nearly inverting their order.

A collection of data by Mr. C. E. Emery, as reported to the American Society of Civil Engineers, gave the following

* Milling Engineer, 1888.

figures, which may be added to those already given in Vol. I.:*

Engine.	Nom. H. P.	Cost.	Effic. of Machine.	I. H. P.	Water per I. H. P. per hr.	Coal.	Water / Coal.	Costs per H. P., 309 days.
Upright portable.....	10	$645	0.80	12.5	38.	5.10	7.5	$109.96
Horizontal "	20	1,981	0.85	23.5	34.	4.25	8.0	73.28
" "	25	2,441	0.86	29.1	32.	4.00	8.0	67.28
Stationery non-cond..	50	5,331	0.88	56.8	27.	3.27	8.25	52.15
" condens. .	100	9,207	0.89	112.4	23.	2.61	8.8	36 02
" "	300	23,899	0.905	331.5	22.2	2.52	8.8	26.82
" "	500	36,220	0.905	552.5	22.2	2.52	8.8	25.66

The following illustrates the advantage of making up heavy trains and employing heavy engines, as well as of adopting a good steam-distribution. The first of these engines has the common Stephenson link-motion; the others the gear of Mr. A. J. Stevens, M. M. of the Southern Pacific R. R., which drives the valve by the combination of eccentric and piston movements. No. 1 has 10 wheels, 6-coupled drivers, 40 tons weight; No. 2 has 12 wheels, 8-coupled, and weighs 61½ tons; while No. 3 has 14 wheels, 10-coupled, and weighs 77 tons,—all exclusive of tender. The last-mentioned was the heaviest and most powerful engine then in the world.†

No.	Cyls.	Wt. train.	Miles per 1 ton coal.	Total cost.	Cost per ton-mile.
1.	18″ × 24″	220.25 T.	20.77	$0.4678	$0.002124
2.	20″ × 30″	378.58 T.	16.88	.5345	.001412
3.	21″ × 36″	595.33 T.	13.81	.6351	.001067

* Transactions, Nov. 1883. See also Wellington on Railway Location, p. 531.

† A train consisting of 156 loaded cars and two cabooses was hauled over the Mississippi Valley R. R. in December, 1885. The load consisted of—

	Pounds.
Cotton.......................................	3,226,000
Cars.......................................	3,239,000
Engine and tender.......................................	147,000
	6,612,000

Compound Engines of 90 tons weight are now built (1891).

Such a study of occasional data has been found by the Author, in a number of cases, to indicate the costs to increase, within practicable limits, nearly as the square root of the weight of train.

Mr. C. T. Main has studied the costs of engine when a portion or all of the exhaust-steam is used for heating purposes.*

The fuel per horse-power per hour is shown in the table. The total consumption is the total burned, and is to be charged to power if no steam is used for heating purposes. The net consumption per I. H. P. per hour is the amount charged to power after deducting the amount of exhaust-steam used for heating purposes.

The conditions are assumed as follows: The compound engine; 100 lbs. initial pressure above the atmosphere; the receiver-pressure to be 5 lbs; the condensing engine initial pressure 80 lbs., and if a portion is high pressure, to exhaust

COAL-CONSUMPTION.

Engine.	Compound.		Condensing.				High-pressure.	
Per cent of exhaust-steam used for heating purposes.	Lbs. of coal per I. H. P. per hour, all coal charged to power.	Lbs. of coal per I H.P. per hour, deducting amount equivalent to exhaust - steam used.	Lbs. of coal per I.H.P. per hour, all coal charged to power.	Lbs. of coal per I.H.P. per hour, deducting amount equivalent to exhaust - steam used.	Do.		Lbs. of coal per I.H.P. per hour, all coal charged to power.	Lbs. of coal per I.H.P. per hour, deducting amount equivalent to exhaust - steam used.
0	1.75	1.75	2.50	Full-condensing, 2.50	Cond'g	2.62	3.00	3.00
10	1.88	1.65	2.55		+	2.40		2.78
20	2.00	1.55	2.60	¾-condensing,	Cond'g	2.17		2.55
25	2.06	1.50	2.63	2.06		2.19		2.44
30	2.13	1.45	2.65		+	2 08		2.33
40	2.25	1.35	2.70	½-condensing,	Cond'g	1.85	Constant.	2.10
50	2.38	1.25	2.75	1.63		1.75		1.88
60	2.50	1.15	2.80		+	1.53		1.65
70	2.63	1.05	2.85	¼-condensing,	Cond'g	1.30		1.43
75	2.69	1.00	2.88	1.19	Full high-pressure	1.31		1.31
80	2.75	.95	2.90	Full high-press.,		1.20		1.20
90	2.88	.85	2.95	0.75		.98		.98
100	3.00	.75	3.00			.75		.75

* Trans. Am. Soc. M. E.; 1888.

against 5 pounds back-pressure; the high-pressure engine with 100 lbs. exhaust against a back-pressure of 5 lbs. The temperature of feed-water is taken at 100° F.

The coal per I. H. P. per hour, when running with steam for power only, is taken as 1¾ lbs. for the compound, 2½ lbs. for the condensing, and 3 lbs. for the high-pressure engines,—fair values for each type.

The sets of data exhibit the method of distribution of costs when using steam for heating and for power.

The conclusions are, that if an amount of exhaust-steam can be constantly used, up to about 80 to 85 per cent of the whole amount exhausted from a high-pressure engine, the most economical plant would be a compound engine; but if more than 80 to 85 per cent of the exhaust could be used for heating purposes, the proper type would be non-condensing.

The *practical* limit of the proportion of exhaust-steam which can be used and yet employ the compound system, when the quantity is variable, is when that proportion requires equal cylinders, and this limit is established by ability to control steam exhausted from the high-pressure cylinder.

The following is Mr. Main's statement of an illustrative example:

CASE I. *Plain Cotton-mill.*—Steam used for running engine and for dressing during the whole year, and for heating the mills for about five months.

Average amount of power required for 1000 spindles on 30's yarn, 18 H. P. Amount of coal required per 1000 spindles for heating and slashing for middle New England about 13 tons. About 30 per cent of this, or 4 tons, is used for slashing, the consumption of which extends through entire year. The remaining 9 tons are used for heating during the five cold months.

4 tons = 8960 lbs. for 308 days = 29.09 lbs. per day or 10 hours = 2.91 lbs. per hour for 7 months.

9 tons = 20,160 lbs. for say 150 days, including Sundays, = 134.4 lbs. per day. About one third of this would be burned when engine was not running, leaving $134.4 \times \frac{2}{3} = 89.6$ lbs. for

10 hours when engine was run, or 8.96 lbs. per hour. $2.91 + 8.96 = 11.87$ lbs. for 5 months.

$$2.91 \times 7 = 20.37$$
$$11.87 \times 5 = 59.35$$

$$79.72 \div 12 = 6.64 \text{ lbs. per hr., average for 12 mos.}$$

As there remains for useful work at exhaust only about 75 per cent of the steam evaporated or admitted to engine, the amount admitted to engine per hour must equal $6.64 \div 75 = 8.85$ lbs.

8.85 lbs. at 2.05 lbs. per H. P. $= 4.32$ average equivalent H. P. of exhaust-steam used for 12 months per 1000 spindles.

$4.28 \div 18 = 24$, or 24 per cent of exhaust-steam is used, leaving 76 per cent to go into low-pressure cylinder. The ratio of areas of low and high-pressure cylinders should then be, for 5 lbs. pressure in receiver $3.5 \times .76 = 2.66$ for 5 lbs. receiver-pressure, or $4.0 \times .76 = 3.04$ for 15 lbs. receiver-pressure.

Oil is an important item of cost. French data give the quantity of oil used in 1865 on compound engines as about 3.5 lbs. per ton of fuel; and this quantity has increased to 6.6 with improvement in engines, but slightly diminished, to one half that quantity, as referred to the horse-power. Common figures are the following: *

	Lbs. per ton coal.
Compound, vertical engines	4.4
" " " with auxiliary engine	4.5
Triple-expansion vertical engines, with auxiliary engine	7.0
" " horizontal " " " "	7.7
Torpedo-boat engines, compound	22
" " " triple	33

In the latter forms, and in high-speed engines generally, when special provision is not made to prevent such waste, much oil is thrown out of the joints by jar and centrifugal force, and an excess is nearly always used to make the machine safe against overheating.

* Revue Maritime et Coloniale; vol. CVIII, 1891.

The coal burned per unit of work done is often taken as the quantity compared in making financial comparisons of the value of steam-machinery. Thus, in marine engineering, the formula

$$V = \frac{(\text{Displacement})^{\frac{2}{3}} \times (\text{Speed})^{3}}{\text{Weight of fuel burned}}$$

is made the measure of the relative value of various constructions. Taking weights in tons, speed in knots, and a time-unit of 24 hours for the latest and best examples of triple-expansion engines

$$V = 15000, \text{ nearly.*}$$

For the compound engine, similarly, the figure is about

$$V = 12000;$$

and for the simple engine of still earlier type, about

$$V = 9000.$$

Otherwise expressed, one ton of coal per day will do the equivalent, in work, of driving $9000^{\frac{3}{2}}$, $12{,}000^{\frac{3}{2}}$, or $15{,}000^{\frac{3}{2}}$ tons, respectively, in these cases, one knot per hour; or, still otherwise, 125 tons of coal, at that conventional standard speed, may transport about 800,000, 130,000, or 185,000 tons of ship and cargo across the Atlantic.

231. Costs of Construction are obtained from the books of the builder, and should exhibit the outgo for materials, labor, wear and tear of machinery, interest on capital invested, and all items in full detail. The following is a very complete statement of the labor and material account in the building of the ordinary standard passenger-engine, as given by the officers of the road for which it was constructed.† To correct the figures for any other time and place, it is only necessary to ascertain the prices of material and labor, and use them as the factors in place of those here taken. This should invariably be

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* See Blechynden on Marine Engineering; Trans. Brit. Inst. Mech. Engrs., 1891.

† National Car-Builder, 1882. Costs in 1891, $7000 and $9000 for 30 and for 60-ton engines.

DETAILED COST OF A PASSENGER-LOCOMOTIVE.

Specifications.		Labor.										Material.			Material and Labor.	
Name.	No.	See Explanation. 1	2	3	4	5	6	Total Hrs.	Average Rate.	Total Cost.	Cost per Lb.	Rough Weight. Lbs.	Total Cost.	Cost per Lb.	Total Cost	Cost per Lb.
Boiler	1	35	71	528	726	978	104	2,442	21.6	$527.33	2.5	21,085	$1,322.37	6.3	$1,849.70	8.8
Grate	2	5		15		30		50	21.0	10.49	1.5	710	14 73	2.1	25 22	3.6
Frame	3	11	22	135	253	67		488	21.5	105.17	2.2	4,866	530.44	10.9	635.61	13.1
Cylinders with bolts and studs	4		30	55	105	365		555	23 7	131.56	2.2	5,896	307.60	5.2	439.16	7.4
Cylinder-heads and casing	5			2	46	37		85	24.4	20.73	1.7	1,219	25.60	2.1	46.33	3.8
Steam-chests, covers, bolts, and studs.	6	5	11	7		54		77	21 5	16.55	2.0	819	19.40	2.4	35.95	4.4
Guides	7		15	30	111	192		348	21.2	73.87	7.4	992	61.98	6.2	135.85	13.6
Guide-yokes.	8			51	35	33		118	20.9	24.63	4.2	585	85.25	14.6	109.88	18.8
Cross-heads.	9		22		58	21		101	21.3	21.44	6.3	338	7.36	2.2	28.80	8.5
Pistons and rods	10	6	10	19	55	35		125	21.2	26.46	3.3	789	27.70	3.5	54.16	6.8
Slide-valves	11	2			21	15		38	21.5	9.18	4.6	198	4.16	2.1	13.34	6.7
Valve-rods	12		1	18	7	26		52	22.0	11.43	10.0	114	1.71	1.5	13.14	11.5
Valve-yokes.	13		2	15	14	14	10	55	24.0	13 46	12.2	110	3.40	3.1	16.86	15.3
Main and parallel rods	14		31	167	36	474		708	22.5	160 26	12.8	1,250	115.13	9.2	275.39	22.0
Driving-wheels, axles, and tires.	15	1		11	8	139	1	160	24.3	38 82	0.3	15,014	632.38	4.2	671.20	4.5
Driving-boxes and cellars	16			24	24	95		143	23.1	33.08	3.9	852	45.88	5.4	78.96	9.3
Equalizers	17		7	67	18	38	37	167	21.3	37.90	8.0	470	7.10	1.5	45 00	9.5
Spring-hangers	18			19		38		57	23.2	13.21	8.3	160	3.06	1.9	16.27	10.2
Driving-springs	19			24		24		48	26.3	12.11	3.0	405	20.13	5.0	32.24	8 0
Driving-spring saddles	20				1			1	17.0	.17	0.1	156	18.72	12.0	18.89	12.1
Pedestal-wedges	21	2	1	17	1			21	15.5	3.24	1.5	215	4.49	2.1	7.73	3.6
Pedestal-shoes	22			10	1			11	16.0	1.74	1.0	168	3.52	2.1	5.26	3.1
Expansion-plates and braces	23	19	2	110	1	153	27	312	21.9	68.20	11.8	579	10.52	1.8	78.72	13.6
Rocker-boxes	24		10	8		38		56	23.4	12.92	3.1	424	8.89	2.1	21.81	5.2
Rocker-arms	25			7		79		86	25.4	21.88	10.2	215	25.80	12.0	47.68	22.2
Links	26			137	48	222	19	426	25.5	108.68	38.8	280	17.15	6.1	125.83	44.9
Eccentric-rods	27			61	1	48		110	21.8	23.94	10.4	230	4.55	2.0	28.49	12.4
Eccentric-straps	28			18	5	71		94	23.6	22.21	4.3	515	10.82	2.1	33 03	6.4
Eccentrics.	29	3	7	14	8	25		57	21.3	12.02	2.2	548	11.49	2.1	23.51	4.3
Tumbling-shaft.	30			9	9	67		85	24.2	20.53	9.5	215	3.23	1.5	23.76	11.0
Tumbling-shaft and spring-boxes	31	5	6	1	3	12		27	19.0	5.12	3.2	161	3.36	2.1	4.48	5.3
Reach-rods	32			15		50		65	23.1	15.04	15.5	97	1.94	2.0	16.98	17 5
Reverse-lever and spring	33		1	33		88	4	126	23.7	29.06	18.4	158	6.40	4.0	35.46	22.4

Sector	34	...	...	3	3	13	...	19	23.5	4.45	21.2	21	.42	2.0	4.87	23.2
Mud-ring and cap	35	6	2	2	4	56	...	70	22.7	15.90	7.5	212	22.73	10.7	38.63	18.2
Throttle-box gland and lever	36	1	...	5	...	36	...	42	24.1	10.13	35.0	29	3.74	13.0	13.87	48.0
Whistle and lever	37	...	...	3	1	8	...	12	23.7	2.84	6.4	44	6.55	15.0	9.39	21.4
Sand-box	38	3	5	1	1	65	...	75	24.0	18.00	4.5	398	11.53	2.9	29.53	7.4
Checks, complete	39	...	...	11	...	38	...	49	23.0	11.28	6.4	176	20.03	11.4	31.31	17.8
Pop-valve, complete	40	...	...	3	2	5	...	10	21.5	2.15	7.2	30	6.95	23.1	9.10	30.3
Dome	41	2	...	6	...	39	...	47	23.7	11.15	1.0	1,149	24.24	2.1	35.39	3.1
Stand-pipe	42	...	1	3	3	...	...	7	20.0	1.40	0.7	202	4.24	2.1	5.64	2.8
Dry-pipe	43	...	...	3	8	80	2	93	24.6	22.87	6.4	359	41.29	11.4	64.16	17.8
Crotch and steam-pipes	44	...	...	8	10	30	...	48	23.2	11.20	2.2	491	10.30	2.1	21.50	4.3
Exhaust pipes	45	1	...	2	...	1	...	4	15.0	.60	0.6	95	2.98	3.1	3.58	3.7
Front end of smoke-box	46	...	5	...	1	13	...	19	21.9	4.15	0.9	438	9.19	2.1	13.34	3.0
Headlight boards and brackets	47	..	3	2	1	13	...	19	22.0	4.18	2.7	158	3.19	2.0	7.37	4.7
Headlight	48	...	...	...	...	...	...	...	...	...	...	...	70.00	...	70.00	...
Running-board and hand-rail brackets	49	...	...	38	1	43	...	82	21.4	17.54	6.1	287	5.74	2.0	23.28	8.1
Jackets and lagging	50	...	...	...	30	40	...	70	26.9	18.80	2.1	888	29.49	3.3	48.29	5.4
Cylinder casing and lagging	51	...	...	2	1	44	...	47	24.5	11.55	26.3	44	2.64	6.0	14.19	32.3
Smoke-stack	52	30	5	11	...	50	8	104	19.0	19.58	2.9	668	31.18	4.7	50.76	7.6
Cab	53	...	...	...	122	6	151	279	27.4	76.35	...	...	36.58	...	112.93	...
Engine-truck	54	...	...	157	12	87	73	329	22.6	74.49	1.0	7,503	195.92	2.6	270.41	3.6
Tender-trucks	55	...	25	119	26	72	...	242	22.6	54.70	0.5	11,729	286.87	2.4	341.57	2.9
Tender-frame (wood and iron)	56	5	...	108	68	116	...	297	21.4	63.44	0.8	7,563	131.64	1.8	195.08	2.6
Tank (steel)	57	7	14	110	143	189	21	484	21.3	103.11	1.8	5,673	300.00	5.5	412.11	7.3
Steam-gauge stand	58	...	...	6	...	114	...	120	24.7	29.67	2.8	106	19.08	1.8	48.75	4.6
Deck-casting	59	...	7	11	8	79	...	105	24.2	25.43	2.0	1,323	27.75	2.1	53.18	4.1
Driving-wheel-covers	60	4	4	83	...	22	...	113	16.5	18.63	7.2	258	7.23	2.8	25.86	10.0
Number-plate	61	1	...	1	1	2	...	5	21.8	1.09	4.3	25	2.86	11.5	3.95	15.8
Pilot	62	5	5	27	80	25	8	150	22.6	33.91	2.2	1,571	16.30	1.0	50.21	3.2
Front and back braces	63	...	...	50	...	81	...	131	21.9	28.68	8.1	353	6.25	1.8	34.93	9.9
Injectors	64	...	...	...	...	...	...	...	...	...	...	...	160.00	...	160.00	...
Injector-fittings	65	...	...	1	...	82	...	83	26.3	21.85	10.2	214	13.28	6.2	35.13	16.4
Chafing-plate	66	1	1	4	...	5	...	11	18.5	2.04	1.9	107	2.24	2.1	4.28	4.0
Total		160	326	2,407	2,120	4,957	489	10,460	...	$2,353.50	...	96,947	$4,889.69	...	$7,241.28	...
Averages		...	...	...	...	...	...	...	22.5	...	22.5	...	...	5.0	...	7.5

done. All figures published here or elsewhere should be taken only provisionally.

Columns 1 to 6 show the hours of work for wages per day of ten hours as follow:

Column 1: wages at 50 and 75 cents.
Column 2: at $1.00, $1.12, $1.20, $1.30, and $1.45.
Column 3: at $1.50, $1.60, $1.65, $1.70, $1.75, and $1.90.
Column 4: at $2.10, $2.15, $2.20, $2.30, and $2.40.
Column 5: at $2.50, $2.70, and $2.90; and
Column 6: at $3.00, $3.30, $3.35, $3.50, and $3.65.

The cost of "erecting" is $468.21, and includes 1857 hours at an average of 25.2 cents per hour. This is distributed in the table, giving to each specification its appropriate part. The average rate per pound given is cents and tenths of a cent, and is computed upon the rough weight of the material. The total cost per pound of the finish weight of the engine complete (which is about 80,000 lbs. empty) is—for labor, 2.9 cents; material, 6.1; and total, 9 cents. The slabs of the frame, received from another shop, weighed 4188 lbs., and are charged in material at $519.73, which is 12.4 cents per pound for labor and material.

Taking the average engine-mileage at 25,000 miles and the fuel at 30 pounds, the total annual expenditure is 335 tons per annum. A saving of 20 per cent would amount to 67 tons, and at $1.50 per ton to about $100 a year,—the interest at 5 per cent of $2000,—and would make the construction of the compound engine decidedly advantageous on the score of financial gain. If the engine can be made to give a mileage of 50,000, the difference in allowable cost, or the gain at the same cost, will represent a capital account of $4000.

The following is the result of an examination into this class of costs in stationary-engine construction.*

The division of labor in a large shop building steam-engines is as follows, percentages being given of the whole number employed in each class of work:

* Mechanics; July, 1887.

	Per cent.
Vise-hands	0.080
Other machinists	0.335
Moulders and core-makers	0.127
Laborers in foundry	0.075
Other common labor	0.125
Boiler-makers	0.090
Blacksmiths	0.056
Wood-workers	0.052
Office-work time-keeping, etc	0.060
	1.000

Nearly 80 per cent of the labor in the machine-shop is skilled and high-priced ; 63 per cent in the foundry is skilled.

Assigning common and clerical labor to the several departments, the distribution of labor is about as follows :

	Per cent.
Machinery, fitting and assembling	50
Foundry-work	21
Boiler-work	16
Blacksmithing	7
Wood-working	6
	100

Another shop shows the following :

Machinists	35
Moulders	10
Core-makers	4
Laborers in foundry	17½
Other common labor	9
Blacksmiths	4
Wood-workers	10
Pattern-makers	2½
Office, watchmen, and teaming	8
	100

In three shops the distribution is :

	Per cent.	Per cent.	Per cent.
Machine-shop	36	34	37
Foundry	30	23	21
Boiler-work	16	23	25
Blacksmithing	7	9	10
Office-work, etc	7	7	5
Pattern-making	4	4	2

Skilled labor in the foundry ranges from one half to three fourths.

In shops devoted to the manufacture of boilers we may estimate about 50 per cent for the skilled boiler-making crafts; 20 per cent for laborers and helpers in boiler-making; 20 per cent foundry-work, and 10 per cent for blacksmithing, threading stay-bolts, and other work. In boiler-making proper—that is, the working of sheet steel and iron—we may for heavy work estimate 54 per cent riveting and calking, 18 per cent flange-turning and the most skilled work, and 28 per cent rivet-heating and helping.

In 1890 the cost of steam-plants in New England cotton-mills, for compound engines and about 500 H. P., was about \$65 per horse-power. Taking the *fixed expenses* at 4 per cent on engine, 5 per cent on boilers, and 2 per cent on other portions, repairs at 2 per cent, interest at 5 per cent, taxes at $1\frac{1}{2}$ per cent on $\frac{3}{4}$ cost, and insurance at $\frac{1}{2}$ per cent on exposed portion, the total average per cent is about $\$65 \times .12\frac{1}{2} = \$8.13$, and the total cost per horse-power \$21.80.

The cost of water-power at Lawrence, as an example, was, according to Mr. Main, about \$19.13, producing an advantage in its favor, in that locality, of about \$2.67, and for the whole 10,000 horse-power of fall the interest at 5 per cent of above \$500,000.

Manual-labor costs are, in a normal, a natural, state of the markets compared, usually a fair gauge, but not an exact measure, of real value. It is probable that, as a rule, under such conditions as ordinarily exist at the given place, the high-priced, and hence highly skilled, labor is worth more and the low-priced, and inevitably unskilled and unreliable, labor is worth rather less than market rates; although a tolerably exact apportionment of brain and brawn to suitable tasks in all

respects is constantly going on all over the world, so far as it is not impeded by artificially imposed restraining circumstances, as by legislation, by local conspiracy to raise or depress the market, and by "strikes" or "lockouts." It was the observation of Mr. Brassey, the most extensive contractor the world had seen at his time, that, where native labor is fairly plentiful, differences in the market prices of labor had little influence on the final cost of the work.* Mr. Burge, however, shows that by properly handling the cheaper labor of new countries, at least, large economies may sometimes be effected ; basing his statement on experience with all classes, from English to Hottentot, and French to Hindoos.†

232. **Repairs and Renewals of Parts** of engines and machinery constitute often a heavy tax on the business. Their magnitude is, however, very variable with differences of kind of work and method of operation. The slow-moving pumping-engine is usually much less costly in this respect than the high-speed stationary or the locomotive engine. The best statistics of this kind are found in railroad reports, and may serve as an illustration of the character and extent of the accounts which should be kept. As a rule, with all well-managed roads and establishments this item will be found to be continually diminishing, as methods and materials of construction and principles and methods of management are improved. Increase constantly going on in the amount of work demanded of the machinery tends to produce the opposite change. Good illustrations may be found in the table: ‡

Later and, to the engineer, more familiar figures are the following, from the books of the Pennsylvania Railway for 1881–4: §

Date.	No. Engines.	p. c. in shop.	Mileage. Millions.	Lbs. coal per m.	Costs per mile. Repairs.	Fuel.	Total.
1880	627	12.5	17.24	77.9	7.29	5.14	13
2	693	10.7	21.00	82.8	6.50	5.39	12.73
4	797	19.2	21.49	85.7	6.97	5.09	12.77

* Life of Brassey; A. Helps, 1874.

† Proc. Brit. Inst. C. E., No. 2012, 1884.

‡ Proc. Brit. Inst. C. E., 1881.

§ See Wellington's Railway Location for a large mass of similar and other valuable data relating to finance.

MATERIALS AND WAGES; REPAIRING ENGINES; JAN. 1876 TO DEC. 1880.

	Passenger-engines.				Goods-engines.	
Number of engines	10		36		6	
How coupled	Driving and trailing		Driving and trailing		Six-coupled.	
Class	Standard No. 1.		Standard No. 2.		Bogie goods.	
Diameter of wheels	6 feet and 6 feet 6 inches.		5 feet 6 inches & 5 feet 8 inches.		4 feet 6 inches.	
Engine-miles run in this period	1,217,122¼		3,861,869½		216.686½	
Average life per engine in miles	279,117		194,106		36,114	
	Amount.	Pence per mile.	Amount.	Pence per mile.	Amount.	Pence per mile.
	£ s. d.	d.	£ s. d.	d.	£ s. d.	d.
A Copper plates	645 5 2	.127	136 11 10	.009	....	....
" stays	163 9 8	.032	65 5 9	.004	....	....
Tubes	855 13 7	.169	159 7 2	.010	... .	...
Fire-bars and brick arch	292 19 1	.057	763 2 3	.047	50 11 6	.056
Sundries for boiler	124 15 11	.025	99 6 7	.006	6 8 6	.007
B Sundries for smoke-box and platform	85 11 1	.017	130 7 3	.008	5 2 5	.006
C Axles	51 8 3	.011	....	....	....	....
Tires	166 17 2	.033	497 16 9	.031	....	....
Springs	284 0 11	.055	548 8 4	.034	21 19 10	.024
Sundries for wheels and frames	63 5 8	.013	133 13 3	.009	16 3 11	.018
D Axle-boxes	181 15 9	.036	589 9 3	.037	33 17 3	.038
" brasses	149 10 7	.030	207 8 4	.013	12 10 4	.014
Big-end brasses	14 6 5	.003	26 3 5	.002	1 9 4	.001
Cylinders and covers	305 18 3	.060	226 16 9	.014	....	.. .
Pistons and sundries for cylinders	20 12 10	.004	109 17 5	.007	2 9 2	.002
Glands and bushes	76 8 0	.015	178 1 1	.010	6 18 9	.008
Slide-valves	68 13 5	.014	187 17 10	.012	9 6 3	.010
Eccentric-liners	105 17 11	.021	174 5 10	.010	6 5 4	.007
White metal	68 10 7	.012	166 11 5	.010	0 19 8	.001
Sundries for machinery	967 10 2	.190	1,905 11 10	.119	38 17 10	.043
E Sundries for mountings	461 15 9	.092	379 4 11	.023	40 13 3	.046
F Sundries for clothing, etc.	155 14 0	.031	320 8 0	.020	31 8 0	.034
Total materials	5,310 0 2	1.047	7,005 15 3	.435	285 1 4	.315
Credits	1,537 12 3	.303	1,049 16 8	.065	34 19 8	.039
Net materials	3,772 7 11	.744	5,955 18 7	.370	250 1 8	.276
Wages: running-shed repairs	1,365 5 9	.269	2,688 14 2	.166	185 2 0	.205
" shop repairs	2,920 3 3	.575	5,809 17 4	.361	165 7 4	.183
Total repairs	8,057 16 11	1.588	14,454 10 1	.897	600 11 0	.664

Fig. 216 is the graphical illustration of these figures. Contemporary figures are always to be employed.

Costs for repairs, with well-built and well-handled machinery, may be reduced to a very small figure. Builders of a well-known type of stationary engine give, for example, less than 2 per cent for average repairs; and the constructors of an

equally well-known boiler state their average at about one fourth of one per cent for minor repairs, and exclusive of renewals.

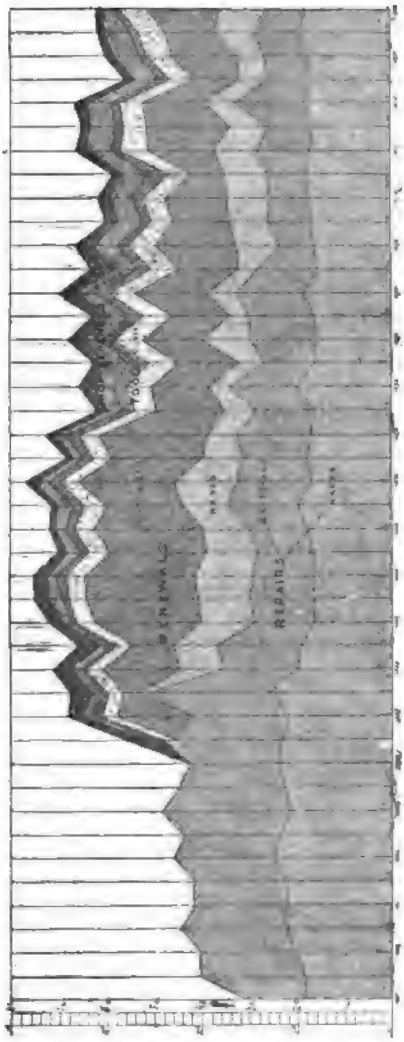

FIG. 216.—REPAIRS AND RENEWALS.

233. Costs of Operation are seldom reported fully, except in railroad work, and occasionally in steam-pumping for cities.

The following figures illustrate well the nature, and give a general idea of the magnitude, of these expenses. Like all such data, however, they must always be revised for the date and location for which a correct account is sought. Some of these data have just been given.

The following is a fair example of good railway work between 1880 and 1888 on a smooth track:

OPERATING COSTS.

Cylinders.	No. of Drivers.	Size of Drivers.	Mileage.			Averages.							
			Passenger.	Freight.	Total.	Pounds Coal per mile.	Quarts Lubricating Oil per 100 miles.	Pounds Tallow per 100 miles.	Pounds Waste per 100 miles.	Passenger. Average Train.	Passenger. Coal per Car-mile.	Freight. Aver. Train on loaded basis.	Freight. Coal per Car-mile.
17 × 24	4	60	5871		5871	48.5	4.1	3.2	1.1	7.2	6.7		
16 × 24	4	60	1965	1834	3799	57.0	1.8	3.3	1.1	6 5	8.8	31.6	1 8
			5764	262	6026	48.5	3.9	3.7	1.1	6.9	7.0	28.8	1.7
			6026		6026	43.5	3.1	3.2	0.9	6.8	6.4		
			4001	356	4357	57.9	5.3	5.2	1.5	7.0	8.3	25.0	2.3

The locomotive is expected to develop about 30 horse-power per square foot of its grate, the engine demanding 25 to 28 pounds of water and 3 to 4 or 5 pounds of good coal per horse-power per hour, and exerting usually from 800 to 1000 pounds tractive effort, as a maximum. At high speeds this .is not required or approached, this limit being attained about 20 or 25 miles per hour. Common standard engines are here assumed. From 50 per cent of the power developed at high speeds to 75 per cent at low speeds is applied to the train, the remainder being lost in the engine and its tender.

Costs of railway transportation in the United States have been enormously reduced in the decade 1880–90, and promise to be still further lessened in the decade 1890–1900. This has come mainly of the increase of loads transported both in the single vehicle and in the train, with corresponding reduction of relative cost of expenditure for materials, and especially for labor. The improved permanent way, and growth in size of engines and in weight of load carried per pound of train carrying it, are the main causes of this gain. In these respects even the best work in Great Britain is exceeded. In

the United States one engine suffices for five miles of road, in Great Britain for only 1.25 miles. The tariff has been reduced, by thus increasing the work done by the rolling-stock and accessory economies, from 2½ cents per ton per mile, on good roads in 1865 to less than one cent in 1890; the actual direct cost of the work being about one half this figure. The annual train-mileage in the United States is not far from 25,000 miles and 15,000 to 18,000 in Great Britain, and the earnings one third greater in the former.*

In 1890, according to Wellington, costs of locomotive operation had fallen somewhat per mile; and as the loads had been continuously increasing in preceding years, the actual net cost per ton-mile of load transported had decreased in much greater proportion. The following are figures given for one of the principal Middle States roads: †

		1884.	1886.	1888.	1890.
Miles per engine		33,548	33,853	36,068	40,714
Costs in cents per mile.	Miscellaneous material	0.11	0.08	0.14	0.18
	Waste and lubricants	0.35	0.21	0.27	0.28
	Fuel	5.34	4.47	6.02	5.72
	Wages	7.46	7.42	7.96	7.04
	Repairs	5.46	5.04	5.63	4.70
	Total costs	18.72	17.22	20.02	17.92

The engine-mileage is double that reported for the same dates for British roads. The figures for fuel are equivalent to a reduction of probably one half per ton per mile in ten or fifteen years.

According to Wellington, the life of the American locomotive may be taken to average 700,000 miles; English engines, 450,000 to 500,000; boilers, 350,000 to 450,000; steel fire-boxes, 120,000 miles to 300,000; copper, 75,000 to 160,000; tubes, 200,000; cranked axles 150,000, plain 300,000; bearings ($\frac{1}{16}$-inch wear), 30,000 to 60,000; tires, 100,000 to 200,000; balanced valves, 30,000—unbalanced, 75,000 to 100,000 miles.‡

* London Engineering, Oct. 2, 1891.

† Engineering News; May 16, 1891.

‡ Location of Railways.

Costs of repairs divide themselves among the various items about as follows:

Parts.	Per cent.
Boiler	20
Running-gear	20
Machinery	30
Fittings	12
Smoke-box	5
Tender	4
" running-gear	9
	100

DETAILS OF EXPENSES of railroad operation as classified is below by engineers and motive-power superintendents, and the arrangement here given, with its figures, are those of Mr. Wellington for a fair average in the United States.*

Trains, 47.0 per cent.	Engines, 18 per cent.	Road Engines.	Fuel	7.6
			Water	.4
			Oil	.8
			Engine-repair	5.6
				14.4
		Switching-engines		3.6
	Train-costs, 17 per cent.	Switching-engines, wages		1.6
		Train-costs, 15.4 per cent.	Engine-wages	6.4
			Car- "	8.5
			" supplies	0.5
				15.4
	Cars.	Repairs, etc.		10.0
		Mileage		2.0
				12.0

* Railway Location; p. 179.

Maintenance of Way, ~~15~~ 23 per cent.	Tracks.	Rail-renewals	2.0
		Track-adjustment	6.0
			8.0
	Road-bed, 7 per cent.	Renewing ties	3.0
		Earth-work, etc	4.0
			7.0
	Yard, etc., 8.5 per cent.	Switches, etc	2.5
		Bridges	3.5
		Buildings	2.0
			8.0
General expenses			30.0

As an illustration of the costs of repairs and preservation in the case of a well-built pumping-engine, take the following, as reported for a Gaskill engine of 1,500,000 gallons capacity, under 100 feet head, by the Engineers' Department of Valparaiso, Ind., for the years 1886—the date of installation—to 1890, inclusive:

COSTS OF REPAIR—PUMPING-ENGINE.

1886.		——	
1887.		——	
1888.		——	
1889.	$2\frac{1}{8}$ lbs. air-pump valves	$ 2 66	
	$9\frac{1}{8}$ lbs. air-pump valves	14 25	
	2 sheets brass, 39 lbs. at 34 cents	13 26	
	15 hours time at 60 cents	9 00	
		——	$39 17
1890.	Lathe-work, 3 hours	1 50	
	2 sheets brass	14 33	
	9 hours at 60 cents	5 40	
	$1\frac{1}{2}$ hours at 40 cents	60	
		——	21 83

Total amount paid for repairs on Gaskill engine	$61 00
Less air-pump valves on hand unused	10 50
Actual repairs 5 years ending January 1, 1891	$50 50

The items for brass were necessary to repair damage from bad oil.

We give below the cost of packings and oil for the years 1889 and 1890:

Packing	$19 93
Oil	35 75
Waste	20 37
Polish	95
	$77 00

Omitting the two items for brass, $13.26 and $14.33, the actual cost of legitimate repairs for five years is $22.91.

234. **The Total Expense of Motive Power,** as presented in the engineer's estimates, must, as already stated, always include the interest, insurance, taxes, and all other incidental costs. The following estimates, communicated to the Author by Mr. Emery, well illustrate the system on which such accounts are to be constructed. They include repairs and renewals and all operating costs for 309 working-days in each year. The figures are partly due to Mr. Hoadley, and such are revised by the author of the statement. It is assumed that replacement must be made at intervals of thirty years, making the present charge on this account, at 6 per cent interest, $21.0814, which sum will provide a total in that time, at compound interest, of $121.08, yielding the required $100 and $21.08 to continue the operation. The friction of engine is taken at $9\frac{1}{2}$ to 20 per cent, according to size of engine. An evaporation at the boilers of 8.8 times its own weight of water is assumed for the value of the fuel. The total weight of coal per horse-power and per hour is assumed at $2.83\frac{1}{3}$

COSTS OF ENGINES AND BOILERS AND EXPENSE OF OPERATION.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Estimated Original Costs.				Data for calculating Operating Expenses.							
Dynamometric Horse-power. Power of Engine.	Kind of Engine.	Cost of Engine and Boiler and all appurtenances, set up in Fall River, Mass., on the first day of January, 1874, including foundations, boiler-settings, pipes, gauges, tools, etc., not including engine and boiler house and chimney.	Cost of Engine- and Boiler-house and Chimney.	Present value of the cost of renewing the Engine, Boiler, and all appurtenances every thirty years. 21.0814 per ct. of Col. 3.	Total, including Cost of Engine and Boiler and Investment for Renewal, excluding Buildings and Chimneys. Sum of Col. 3 & 5.	Percentage of Friction of Engine.	Indicated Horse-power.	Feed-water per Indicated Horse-power per hour.	Feed-water evaporated in Boilers per Pound of Coal.	Coal per Indicated Horse-power per hour.	Total Coal per Day of 10 hours, with ⅛ added for starting and banking fires.	Insurance—Yearly Cost at the Rate of ½ of 1 per cent on total valuation in Col. 3.	Taxation—Yearly Cost at the Rate of $15 per M, on 75 per cent of total amount in Col. 3.
H. P.		Dollars.	Dollars.	Dollars.	Dollars.	Per ct.	1 H. P.	Lbs.	Lbs.	Lbs.	Lbs.	Dollars.	Dollars.
5	Portable upright.......	645	313	135.98	781	20	6.25	42	7.5	5.60	394	3.23	9.68
10	" "	988	408	208.28	1,196	20	12.50	38	7.5	5.10	717	4.94	14.82
15	" "	1,487	504	313.48	1,800	18	18.29	36	7.5	4.80	988	7.44	22.31
20	" horizontal.........	1,981	589	417.62	2,399	15	23.53	34	8.0	4.25	1,125	9.91	29.72
25	" "	2,441	699	514.60	2,955	14	29.07	32	8.0	4.00	1,308	12.21	36.62
50	Stationary, non-condensing..	5,331	1,029	1,123.85	6,455	12	56.82	27	8.25	3.27	2,091	26.66	79.97
100	Condensing, single	9,207	1,487	1,940.96	11,148	11	112.36	23	8.8	2.61	3,300	46.04	138.11
150	" "	13,046	2,129	2,750.28	15,796	10	166.67	22.2	8.8	2.52	4,725	65.23	195.69
200	" "	16,785	2,745	3,538.51	20,324	9.5	220.99	22.2	8.8	2.52	6,265	83.93	251.78
250	" "	20,426	3,304	4,306.09	24,732	9.5	276.24	22.2	8.8	2.52	7,831	102.13	306.39
300	" "	23,899	3,841	5,038.24	28,937	9.5	331.49	22.2	8.8	2.52	9,398	119.50	358.49
400	" "	29,958	5,722	6,315.57	36,274	9.5	441.99	22.2	8.8	2.52	12,530	149.79	449.37
600	" "	36,220	7,260	7,635.68	43,856	9.5	552.49	22.2	8.8	2.52	15,663	181.10	543.30

COSTS OF ENGINES AND BOILERS AND EXPENSE OF OPERATION—*Continued.*

15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Operating and other Current Expenses.												Capitaliza'n at 6 p.c.		Sum of Original Cost and Capitalization of Operating and other Expenses, including Investment for Renewal.	Total Cost per Horse-power maintained continuously, including Proportion of Original Cost of Plant, Renewals, etc.	
Engineer.		Firemen.		Supplies, Oil, Waste, Packing, etc.		Repairs—Ordinary and Extraordinary—including Grates and Flues.		Total Cost of Operating and other Current Expenses, except Coal.		Cost of Coal at $4.17 per Ton, including 25 Cents per Ton for Carting.		Of Costs as per Col. 23, of Operating and other Current Expenses, except Coal.	Of Cost of Coal.			
Per Day.	Per Year of 309 Days.	Per Day.	Per Year of 309 Days.	Per Day.	Per Year of 309 Days.	Per Day.	Per Year of 309 Days.	Per Year. Sum of Cols. 13, 14, 16, 18, 20, and 22.	Per Horse-power per Year.	Per Year of 309 Days.	Per Horse-power per Year.			Sum of Cols. 6, 27, and 28.	Excluding Buildings and Chimneys.	Including Buildings and Chimneys.
Dols.	Dols.	Dols	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols.	Dols	Dols.
1.75	540.75			.20	61.80	.13	40.17	655.63	131.13	226.64	45.33	10,927.17	3,777.33	15,485	3,097	3,159
1.75	540.75			.25	77.25	.16	49.44	687.20	68.72	412.44	41.24	11,453.67	6,874.00	19,524	1,952	1,993
2.00	618.00			.27	83.43	.17	52.53	783.71	52.25	568.33	37.89	13,061.83	9,472.17	24,334	1,622	1,655
2.00	618.00			.30	92.70	.22	67.98	818.31	40.92	647.14	32.36	13,638.50	10,785.67	26,823	1,341	1,361
2.25	695.25			.33	101.97	.27	83.43	929.48	37.18	752.41	30.10	15,491.33	12,540.17	30,987	1,240	1,268
2.00	618.00	1.40	432.60	.36	111.24	.44	135.96	1,404.43	28.09	1,202.82	24.06	23,407.17	20,047.00	49,909	998	1,019
2.25	695.25	1.50	463.50	.40	123.60	.77	237.93	1,704.43	17.04	1,808.28	18.98	28,407.17	31,638.00	71,193	712	861
2.50	772.50	1.50	463.50	.47	145.23	1.00	309.00	1,951.15	13.01	2,718.00	18.12	32,519.17	45,300.00	93,615	624	766
2.50	772.50	1.50	463.50	.55	169.95	1.24	383.16	2,124.82	10.62	3,603.86	18.02	35,413.67	60,064.33	115,802	579	716
2.75	849.75	1.50	463.50	.65	200.85	1.47	454.23	2,376.85	9.50	4,504.68	18.02	39,614.17	75,078.00	139,424	558	690
3.00	927.00	1.50	463.50	.80	247.20	1.70	525.30	2,640.99	8.80	5,406.08	18.02	44,016.50	90,101.33	163,055	544	672
3.00	927.00	2.25	695.25	.95	293.55	2.20	679.80	3,194.76	7.99	7,207.72	18.02	53,246.00	120,128.67	209,649	524	667
3.00	927.00	3.00	927.00	1.15	355.35	2.87	886.83	3,820.58	7.64	9,009.94	18.02	63,676.33	150,165.67	257,698	515	660

pounds. Insurance is taken at ½ per cent of the total valuation, and taxes at $15.00 per thousand on a three-fourths valuation. All costs are taken, so far as possible, from actual transactions. In all cases such estimates must be taken as simply illustrative, and contemporary data must be always collected at each location in making such comparisons, whenever accuracy is essential.

The usual rough estimate of the deterioration of machinery is not far from the following:

Engines	5 per cent.
Boilers	10 " "
Machinery	12 " "
Belts, etc.	30 " "

But these figures, with careful management, may be greatly reduced.

"*Subdivided power*" will sometimes be found to give the best results when total expenses are reckoned for alternative plans. As an example of such an arrangement we have the following:

The Dunnell Manufacturing Company of Pawtucket, R. I., formerly distributed all power from a large Corliss engine. As later arranged, boiler-power is concentrated at one point, and their 38 engines—a total of 1405 horse-power—are distributed as required. The dye-house contains one engine of 100 horse-power and one of 45 horse-power, the two driving shafting at right angles.

The color-shop contains a 60 horse-power and 25 horse-power engine, on divergent shafting angles, and the bleach-house 125 horse-power. The calender-room is given an engine of 200 horse-power. The generator-room has three engines of 150 horse-power each. From here a circuit is taken into another building to furnish current for 10 motors driving printing-machines. Other generators supply 1500 incandescent lamps. By the use of motors the speed of each printing-machine may be graduated from 25 to 200 revolutions of the pinion-shaft.

The repair-shop is supplied with an independent steam

plant, and as the exhaust cannot be utilized, a 35 horse-power compound upright engine is employed.

235. **Motors Compared** should always be studied on this basis of total costs. Comparing the costs of steam with those of water-power, the following figures are obtained for an Eastern location:

COST OF STEAM AND OF WATER POWER: COTTON-MILL OF 25,000 SPINDLES AND 400 H. P.

Steam.

Interest on cost of plant,	6 per cent on $35,680,	$2,141		per year.
Sinking-fund for renewal,	2½ " " "	892		"
Operating expenses,	10 " " "	3,195		"
Fuel,	20 " " "	7,208		"
	Total	$13,436		
	Per horse-power.......		$33 59	"
			50	per spindle.

Water.

Interest on cost of plant, 6 per cent on $35,680....	$2,141		per year.
Depreciation and repairs, 2½ per cent....	893		"
Water-tender....................................	618		"
Interest on the cost of the steam-plant necessary to supply steam for heating mill, use of slashers, etc., 6 per cent on $5796.............. ...	348		"
Repairs and renewals of same..................	318		"
Fuel...	1,900		"
Labor..	618		"
Total	$6,836		
Per horse-power........		$17 09	"
Difference per horse-power between steam and water,		$16 50	"

In comparing steam and water power, the conditions and environment of each, assuming its adoption determined upon, must be carefully studied. The steam-power may be made what is desired, and located wherever found advisable. Water-power must be taken where it can be found, and is incapable of modification of power, or variability, by the engineer. A low fall demands more expense in dam and wheels than a high

one. An inconveniently accessible fall requires expense in reaching it, and in its utilization. Variable flow leads to increased cost both of "plant" and of operation. Steam-power may be made very inexpensive in cost of fuel when the exhaust-steam can be utilized for heating buildings, kiers, or for other purposes.* Mr. Main considers that the essential points which must be considered are:

(1) Quantity of water during a dry year.

(2) Uniformity of flow during the year, considering the storage capacity, natural and artificial.

(3) Head of fall.

(4) Conditions which fix the expense of building dam and canal, and flowage of land.

(5) Conditions which affect the cost of foundations for buildings.

(6) Climatic conditions which determine the permanency of the falls.

(7) Freight charges for fuel, supplies, raw materials, and finished product.

(8) How much low-pressure steam can be used for heating purposes, and whether exhaust-steam can be used for those purposes.

(9) Is water needed for other purposes than power, and in what quantities?

(10) The social and sanitary conditions which make it possible to procure and keep good help.

(11) The greater uniformity of speed with steam than with water power.

All items except the last two can be easily approximately estimated.

Messrs. Brooks and Steward, comparing other small motors with gas-engines, thus take into account:

(1) The cost of gas or coal consumed.

(2) The cost of water used.

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* Value of water-power; Trans. Am. Soc. M. E., 1891; vol. XIII. No. CCCCLXX.

(3) Lubrication.
(4) The cost of attendance.
(5) Depreciation and repairs.
(6) Interest on capital invested.

(1) The average consumption of gas in a gas engine per effective horse-power per hour, including igniting flames, is about 30 cubic feet.

The consumption of coal per effective horse-power per hour by small steam-engines is about seven pounds.

(2) The water used in the water-jacket of a gas-engine are not taken into the estimate.

(4) A gas-engine requires little or no attendance. A man can accomplish $\frac{5}{6}$ of a day's work and still take full charge.

Steam-engines of this size require from $\frac{1}{2}$ to 1 day's attention.

(5) As regards depreciation, gas and steam engines have about equal terms of life; for, while gas-engines have less complication of working parts than steam-engines, they are subject to more severe and abrupt strains.

(6) Interest will be proportional to the amount of capital invested.

For *intermittent* work the gas-engine is much more economical than the figures indicate; and this fact, together with its characteristic safety, convenience, and cleanliness, give it great advantages. The steady reduction observed in the cost of gas is continually aiding its introduction.

All costs are, however, constantly variable with time and location.

The following shows the cost of a day's run in cases actually thus taken:

GAS-ENGINE, 8 H. P. actual, 10 hours.

(1) 2400 cubic feet gas at $2.50 per 1000	$6 00
(2) Water	0 00
(3) Lubrication	0 20
(4) $\frac{1}{6}$ day's labor at $2.00	0 33

(5) Depreciation, etc., at 12 per cent per year, $\frac{12}{360}$ per cent on \$1075.......................... \$0 36
(6) Interest at 5 per cent per year, $\frac{5}{360}$ p. c. on \$1075.. 0 15

Daily expense.......................... \$7 04

STEAM-ENGINE, 8 H. P. actual, 10 hours.

(1)	Coal, $\frac{560}{2240}$ tons at \$5.00..........................	\$1 25
(2)	Feed-water, 65 cubic feet at \$1.25 per 1000.......	0 08
(3)	Lubrication..............................	0 15
(4)	$\frac{1}{2}$ day's labor at \$2.00..........................	1 00
(5)	Depreciation, etc., at 12 p. c. per year, $\frac{12}{360}$ p. c. on \$800	0 27
(6)	Interest at 5 p. c. per year, $\frac{5}{360}$ p. c. on \$800.......	0 11
	Daily expense..........................	\$2 86

The following figures were given for the expense of running a small hot-air engine in a printing-office in New York City. It uses $4\frac{1}{2}$ pounds of coal per H. P. per hour; every third year relining costs \$100.

HOT-AIR ENGINE, $2\frac{1}{2}$ H. P. actual, 10 hours.

(1)	Coal, $\frac{115}{2240}$ tons at \$5.00..........................	\$0 25
(2)	Water..............................	0 00
(3)	Lubrication..............................	0 10
(4)	Attendance, same as for gas-engine...............	0 33
(5)	Depreciation, etc., at 10 p. c. per year, $\frac{10}{360}$ p. c. on \$750..............................	0 21
(6)	Interest at 5 p. c. per year, $\frac{5}{360}$ p. c. on \$750........	0 10
	Daily expense..........................	\$0 99

The cost of one horse-power per hour is thus:

With gas-engine............................	$8\frac{3}{4}$ cents.
" steam-engine..........................	$3\frac{1}{2}$ "
" hot-air engine..........................	4 "

Where gas can be purchased at $1.25 per thousand cubic feet, Mr. E. M. Jenkins estimates as follows for small engines:*

FOUR H. P. STEAM-ENGINE.

	Per day.
Coal, 500 pounds at $2.50 per ton	$0.625
Labor	.250
Oil, waste, repairs	.125
Water	.055
Insurance	.065
Total	$1 12
Per H. P. per month	$7 28

FOUR H. P. GAS-ENGINE.

	Per month.
Gas per month per H. P	$3 90
Oil and waste " "	60
	$4 50

SEVEN H. P. STEAM-ENGINE.

	Per day.
Coal, 700 pounds	$0.875
Labor	.500
Oil, etc	.150
Water and insurance	.155
Total	$6 24
Cost per month	$43 68
" " " per H. P	6 24

SEVEN H. P. GAS-ENGINE.

	Per month.
Gas per H. P	$3 25
Oil, etc., per H. P	60
Total	$3 85

* Proc. Western Gas Assoc., 1891; Iron Age, June 4, 1891, p. 1068.

The first cost of the gas-engine is given as from 50 to 100 per cent in excess of that of the steam-engine. Where concentration of great power within small volume is necessary, steam must be used. Thus Mr. Thorneycroft has developed 800 horse-power from a water-tube boiler weighing but 10 tons, with an air-pressure in its furnaces of three inches of water. No such results are possible with any other system, and the concentration of power illustrated in the fast transatlantic steamers, elsewhere described, can only be secured by the use of steam.

Where, in the case of mill-engines, steam in considerable quantity is required for bleaching, dyeing, and other purposes, the non-condensing may have an important advantage over the condensing engine. Thus, Mr. Leavitt, comparing such cases at Lowell in 1877, found the following costs:*

COST PER DAY OF 10 HOURS PER I. H. P.

Condensing engine	10.2 cents.
Non-condensing engine	2.55 "

Equivalent to $30.00 per year of 3000 working hours for the first and $7.65 for the second case, including wages, supplies, and interest on a cost, allowing for labor the difference between that needed with and without the engine in operation. The very great difference here reported is due to use of the exhaust-steam of the non-condensing engines.

Financial considerations often dictate the alteration of an old engine rather than its replacement by a new one. The Author has often, for example, designed a new cylinder to fit an old frame, and thus displaced a cylinder of bad design and incorrect proportions; and an inefficient valve-gear and governor to make way for cylinder, valve-motion and governor well suited to the place and purpose of the machine,—saving a large proportion of the cost of a new engine and securing nearly or quite as good performance, economically and in regulation,

* Trans. Am. Soc. C. E., March 1891, vol. XXIV. No. 467, p. 213.

as a new engine would have given. In some cases simply "speeding up" the old engine and applying a modern form of governor will effect great improvement. Engines may often be easily compounded, either by the McNaught system or something equivalent, or by adding a complete high-pressure engine of proper size, and exhausting into a receiver from which the old low-pressure engine may take steam.

Similarly, improved boiler-setting, the addition of feed-water heaters at the engine, and "economizers" in the chimney-flue may give such economical advantage as may defer the purchase of a new boiler a long time, at a critical period financially.

On shipboard it has often been found practical to compound simple or to convert compound engines into triple, or even into quadruple, expansion engines. In such cases the change is commonly made when the boilers have been so long in use as to make it desirable to replace them, and thus, the the new boilers being designed for the higher pressure, the change is made at minimum cost. It is sometimes quite practicable to leave the existing engine unaltered, placing the new cylinders above the old, and working the valves from the old gear. Old styles of engine are thus often capable of conversion into new at but a fraction of the cost of entirely new machinery, and a vessel incapable of paying expenses may be made fairly profitable. Such alterations become profitable, also, where the low cost of fuel would make it unwise to attempt new constructions.

The financial advantage of compounding existing engines is great where fuel is costly, slight where fuel is cheap. It has been found in some cases in which the comparison was made that the saving would not pay for the cost of alteration in many years; and it may be perfectly possible that cases may arise in which the fuel is so inexpensive that the saving, ordinarily taken at 15 to 20 per cent of the fuel, would not pay even moderate interest on the cost.

A locomotive running 40,000 miles per annum and burning 4 pounds of coal per train-mile, would consume 1,600,000

pounds or 800 "round tons" per year. If the price be $2.50 per ton, the cost of its fuel being $2000, a saving of 15 per cent, assured, would be $300,—the interest at 5 per cent of $6000 expended. In the United States, where the lines are longer, the work of the engine and its working time greater than in Europe, the compound engine should give, on the whole, better returns.

Steam and water power compared on the basis of local and actual costs, will often give rise to conclusions which will be widely different in different places and under different, though usual, conditions. In the Eastern portion of the United States the various charges for water in Manchester, Lowell, and Lawrence, are as follows: *

Manchester.

About $300 per year per mill-power for original purchases.

$2 per day per mill-power for surplus.

Lowell.

About $300 per year per mill-power for original purchases.

$2 per day per mill-power during "back-water."

$4 per day per mill-power for surplus under 40 per cent.

$10 per day per mill-power for surplus over 40 per cent and under 50 per cent.

$20 per day per mill-power for surplus over 50 per cent.

$75 per day per mill-power for any excess over limitation.

Lawrence.

About $300 per year per mill-power for original purchases.

About $1200 per year per mill-power for new leases at present.

$4 per day per mill-power for surplus up to 20 per cent.

$8 per day per mill-power for surplus over 20 and under 50 per cent.

$4 per day per mill-power for surplus over 50 per cent.

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* Mr. C. T. Main; Trans. A. S. M. E., 1889.

One mill-power in Lawrence = 30 cubic feet of water falling through 25 feet head per second. The horse-power of the water is due to a head of about one foot less than 25, to allow for the loss in getting the water on and off the wheel, so that we have

$$1 \text{ M. P.} = \frac{30 \times 24 \times 62.3}{550} = 81.56 \text{ H. P.}$$

With wheels of 80 per cent efficiency the power on wheel-shaft equals 65.25 H. P.

A mill-power in Lowell is equivalent to that in Lawrence.

In Manchester a mill-power = 38 cubic feet per second on a fall of 20 feet. Allowing one foot loss, we have

$$\frac{38 \times 19 \times 62.3}{550} = 81.78 \text{ H. P.}$$

With 80 per cent efficiency on wheels, this equals 65.42 H. P. on wheel-shaft.

The H. P. per M. P. is near enough to call it the same for these three places, 65 H. P. The net effective horse-power would be about 60.

Attendance, one man at $2.00, for 308 days.....	$616
Oil, waste, and supplies.......................	100
	$716

Or about $0.72 per H. P. per year for a 1000 H. P. plant.

As a general rule, the costs of water-power are less than those of steam-power, but the advantages of steady operation often more than balance this difference in cost.

The use of steam has increased more rapidly than that of water-power, and this change will undoubtedly continue for an indefinite period. Nevertheless, the use of water-power is growing, and will continue to grow so rapidly as to furnish an enormous and an extending market until the steady decrease of water-power available throughout the country shall, ere many generations, assume such serious proportions as to

affect the business interests of the nation. The number of water-wheels in use in the United States exceeds the number of steam-engines; and of the total amount of power furnished by the latter the principal part is obtained from a half-dozen large rivers. The amount of power so available is indicated by the fact that the Merrimac River supplies 10,000 horse-power at Manchester, N. H., and an equal amount at Lowell, under heads of about 50 and 35 feet, respectively; the Mohawk is capable of furnishing, in good seasons, 10,000 horse-power at Cohoes, N. Y., under a maximum head of 100 feet; the Androscoggin gives, at Lewiston, Me., an equal amount under 50 feet head, and the Connecticut River at Holyoke, 18,000 horse-power; while the Falls of Niagara will undoubtedly in time be called upon to yield a part at least of their 3,000,000 horse-power with a head of 160 feet.

These great water-powers are usually under the control of a corporation, empowered, by act of the legislature of the State within which the fall is situated, to erect and maintain dams and accessories, and to use or to rent the power so obtained at such rates as they may find most profitable.* Thus it happens that the whole power of the Merrimac at Lowell and that of the Connecticut at Holyoke is controlled by a "Water-power Company" at each place. The water-power companies are usually directed by a policy which encourages the introduction of a variety of manufactures, and the use of the available power to the best advantage. The method of sale of power is substantially the same at all the principal falls. Power is sold by the "mill-power," which is a quantity of slightly variable amount. At Lowell it is the equivalent of 30 feet per second under 25 feet head; at Minneapolis, where the whole current of the Mississippi pours over the Falls of St. Anthony, the mill-power is 30 cubic feet per second, under 22 feet head, or its equivalent; at Holyoke it is the equivalent of 38 cubic feet under a head of 20 feet. The prices are very

* At Lowell the six or seven principal manufacturing corporations form the Water-power Company, owning shares of stock in proportion to the power owned by each company.

variable with location and time of sale. The mill-power at Lowell is equal to about 85.2 theoretical H. P. 16 hours per day. None is sold, all being used by the mills included in the Water-power Company. At Holyoke the payment* is $300 per annum for the equivalent of 90 H. P., or $4.17 per horse-power.

According to Mr. Main, in New England the cost of the plant will vary inversely with the head, as the greater the head the smaller the wheel; but under a head of 30 feet the cost of a modern plant of 1000 H. P. would be approximately as follows:

Feeder head-gates, rack, etc.	$3 70	per N. H. P.
Steel penstocks	14 60	"
Wheel-pits, piers, etc.	11 20	"
Wheels, casings, draught-tubes, and shafting.	22 00	"
Total cost of plant	$51 50	"

To be able to maintain speed at all times, an extra allowance of wheel-power is made, of $33\frac{1}{3}$ per cent, bringing this cost to $\$51.50 \times 1.33\frac{1}{3} = \$68.67$. To this add a sinking-fund for renewals, 4 per cent; repairs, $1\frac{1}{2}$ per cent; general expenses, such as insurance, taxes, interest, etc., 6 per cent.

Summing up:

Sinking-fund	$2 75
Repairs	1 03
General expenses	4 12
Total	$7 90

Wages of a wheelman, at $2.00 per day for three hundred and nine days a year, $618, plus supplies, such as packing, oil, and waste, $100, about $0.72 per H. P. per annum.

* This is not the *price* of the power: it is only that *portion* of the purchase price of land and power which the Water-power Company demands in the form of an annual rental. The balance may be, and generally is, paid in money or notes.

Total cost per N. H. P. per annum:

Cost of water	$14 00
Sinking-fund, etc	7 90
Attendance and supplies	72
	$22 62

If the water is supplied from the surplus at four dollars per mill-power per day, this must be increased by $\frac{4 \times 309}{65} - 14 = 5.01$, making the cost $27.63; and if the water is "surplus" at $2.00, the cost decreases to $16.20.

In a cotton-mill making only white cloth there is ordinarily little use for exhaust-steam; and the compound engine is taken to be the best type.

In woollen-mills, and in cotton-mills making colored goods, there are demands the year around for steam for dyeing and drying, and there will in winter-time be a call for three quarters of the exhaust-steam.

If one half is driven by water-power, the engine should be a simple engine, running against back-pressure.

Consider these cases, and 1000 N. H. P., adopting the same authority.

A well-designed compound engine should, using high steam of 150 lbs. gauge-pressure, have an efficiency of mechanism 93 per cent of the H. P.; therefore the engine should indicate $\frac{1000}{.93} = 1075$ H. P.; but we will take 1100 H. P. The engine is to run ten hours a day, and, allowing for stopping and starting, ten and one quarter hours per day for three hundred and nine days a year. An engine of this type should run on one and three quarter pounds of coal per H. P., including all used for starting and banking, and we take the average cost of such coal at $4.50 per long ton. This brings the cost per H. P. per annum to $12.25, allowing no credit for exhaust-steam.

If the steam taken from the receiver for heating, etc., is one

fourth, we must credit the engine with nine tenths of one fourth cost of coal, which reduces it to $9.49.

Assume an engineer at $3.00, oiler at $1.50, two firemen at $1.50 each, and one coal-passer at $1.20, or an annual pay-roll of $2688.30, $2.44 per H. P. per annum. Engine-room supplies, $250 per annum, or $0.23 per H. P.

Summing up:

PERATING EXPENSES.

Net coal chargeable to engine	$9 49	per H. P.
Attendance	2 44	"
Supplies	23	"
Total running expenses	$12 16	"

COST OF PLANT.

Engine, including piping and foundation	$27.00	per H. P.
Engine-house	5 00	"
Boilers ready for use	10 00	"
Feed-pumps, injectors, etc.	1 50	"
Boiler-house, chimney, and flues	6 00	"
Coal-shed, tracks, etc.	3 00	"
Total	$52 50	

Sinking fund at 5%	$2 62
Repairs 2½%	1 31
General expenses, insurance, taxes, interest, etc., 6%	3 15
Total	$7 08

TOTAL COST PER H.P. PER ANNUM.

Running expenses	$12 16
Charges on plant	7 08
Total	$19 24

The cost per net horse-power per annum will be eleven tenths this, or $21.16, reduced by the proportion of expenses equivalent to the portion of the steam used for heating and slashing.

The other case is where all the exhaust-steam is used at a pressure of ten pounds above the atmosphere, for other purposes.

In such an engine the cost of coal per H. P. is taken as three pounds per hour, charging all to the engine.

If the efficiency of the boiler is 80 per cent, and the engine works between 150 lbs. initial pressure and 10 lbs. back-pressure, it will convert about one tenth of the heat into useful work, or 0.3 of a pound of coal per H. P., which may be increased to 0.5 by cylinder-condensation.

The boiler-plant for such an engine will cost more than for the first engine considered; but this is offset by decreased cost of engine, if the single-cylinder type is chosen. The running expenses will be practically the same, but a larger deduction can justly be made from the cost of power.

The cost of fuel chargeable to power is reduced to $3.50 per H. P. per annum, and the total cost per H. P. per annum to $13.25, and per net H. P. to $14.58.

To sum up, we have the cost per net horse-oower per annum at this point:

Water-power under original leases	$22 62
Surplus water at $5 per M. P. per diem	27 63
" " " $2 per M. P. per diem	16 20
Compound engine, one quarter exhaust, used for heater, etc.	21 16
Single-cylinder, all exhaust used	14 58

In no two localities, however, is it likely that such comparisons will give similar results.

236. Analyses of Costs, as given in the preceding articles, show that some of the items may be taken as constant, and others as variable with those conditions which are themselves

the variables in the comparison of different types and different machines with a view to selection. It is further evident, also, that the number of variable quantities is so great that it would be quite impracticable, in any case, to solve the general problem and to determine precisely what set of mutually dependent variables, independent variables, and constants will give a true minimum of costs and maximum of profits.

Insurance and taxes may be assumed to be substantially the same, whatever constructions and plans may be adopted; in most cases, although not in all, the wages paid will be the same; the expense of fuel will be variable, both with the efficiency of the apparatus and the quality of the fuel itself; costs of oil and miscellaneous supplies will be variable, both with character of machinery and of attendants, and in a manner outside of all rules; repairs can neither be predicted nor estimated for with any confidence or exactness; minor costs will come in in an irregular and uncertain way; and the total cost will thus have a magnitude which can only be roughly estimated and never exactly foreseen. But the principal items are those relating to costs of fuel and of attendance, and the market rates for both can be easily ascertained; while the known character of the machinery—assuming it not to be experimental—affords a means of judging from earlier experiences what may be expected. Such tables as those which have been previously given in illustration will then furnish the only means of comparison and for final judgment and conclusions, and, if prepared by an expert and experienced engineer, will usually yield satisfactory conclusions.

In all cases it is wise to reduce all statements, in this manner, to an estimated total cost for erection and continuous maintenance, as measured by capital entering the investment, capitalizing all costs of operation, maintenance, and renewal on the assumption that the work will require to be done perpetually or indefinitely. As has been seen, this, in the case of the ordinary steam-plant, amounts to \$600 or \$700 per horse-power for large sizes, and up to \$2000 or \$3000 for the smallest powers.

Mr. Fanning estimates the costs of a 500 H. P. milling-plant thus: *

Condensing engine, boilers, etc.		$30,000
Present worth of renewals at ten years, annual payments		22,080
Annual costs of fuel	$50 per H. P.	
" " " supplies	1 85	
" " " attendance	10 25	
" " " repairs	3 00	
" " " insurance, etc.	1 20	
" " " incidentals	75	
Total per H. P.	$67 05	
500 H. P. at $67.05	$33,525	
Annual expense capitalized at 6 per cent		$558,750
Total capitalized cost		$610,870

or $1221.74 per H. P.

It is assumed that the machinery is in operation 24 hours per day, 300 days per year.

For textile manufactures, demanding 1.5 H. P. per employé as a minimum, and flouring-mills, requiring 12 to 15 H.P. per man, water-power may often successfully compete with steam.

237. Problems of Maxima and Minima thus arise in the course of the work of the designing and of the constructing engineer, as well as in the experience of the purchaser and user of motors, which are of very serious importance, and often affect interests of great magnitude. Only tentative methods can be employed, however, as a general rule, in their solution, and only approximate solutions can be secured. But these results, as obtained by a competent practitioner, are often, perhaps commonly, so accurate as to serve as satisfactory guides. In some minor details, occasionally in matters of considerable im-

* Engineering, Oct. 24, 1890, p. 475.

portance, it is found possible to use really exact and reliable processes and to obtain correct results. Thus it is possible, when the working conditions and market rates are specified with precision, to say just how large a steam-boiler should be designed for a given power; what maximum power or maximum profit may be sought from a boiler in place; what power may be demanded of a given engine, or what size of engine may be constructed for a given power. The latter two cases and some similar problems have been elsewhere discussed in this work.* The two problems which follow relating to the boiler are taken from another work by the Author.†

Mr. E. N. Wright thus treats one of the familiar commercial problems in engine-proportioning:—

The total cost of power for any engine is divided into three parts:

(1) Interest, depreciation, and repairs equal some per cent of first cause.

(2) Steam used.

(3) Constant expenses (engineer's wages, oil, waste, etc.).

Let $F =$ first cost of engine;
$P =$ per cent covering interest, depreciation, and repairs;
$S =$ pounds of steam used per year;
$a =$ cost of producing one pound of steam;
$C =$ constant expenses (wages, oil, waste, taxes, insurance, etc.).

Then yearly expense $= PF + aS + C$.

Let a subscript $_a$ make above quantities apply to a given engine A.

Let a subscript $_b$ make above quantities apply to any other engine B.

Then, for comparison of price under equal annual expense, $F_a + aS_a + C_a = P_bF_c + aS_a + C_b$.

Solving for F_a will give the price the customer can afford to pay for A. For instance, compare S with an engine, B,

* Part I., Chap. VII., p. 705 *et seq.*

† Manual of Steam-boilers; N. Y., J. Wiley & Sons.

worth \$2500 $= F_b$. Allow that both may be belted directly to their work, uniform load, cost of space occupied and of erection to be the same.

Assume $P_a = P_b = 0.125$; $C_a = C_b = \$1200$.

All of the above favoring B,

$$F_a = F_b + \frac{a}{0.125}(S_b - S_a).$$

Take 300 days of ten hours each per working year. Coal at \$4.48 per long ton, and boiler evaporation 10 pounds per pound of coal. Then

$$S_b = 28 \text{ lbs.} \times 200 \text{ (H. P.)} \times 10 \times 300;$$
$$S_a = 23 \quad \times 200 \quad \times 10 \times 300.$$

$F_a = \$2500 + \$4800 = \$7300$; while A sells for \$4250.

Had we allowed a counter-shaft for B, this should add to its first cost. Similarly, extra cost of foundation and erection may be added to first costs of B.

The value per year of extra space occupied should add to C_b.

When A saves part of engineer's wages, oil, or waste, the amount should reduce C_a.

If A has less repairs, P_a will be less than P_b. A similar comparison may be made between a single engine and subdivision of power, providing the several factors are known.

Take an example in which $P_a = 0.12$; $P_b = 0.15$; $C_a = \$900$; $C_b = \$1200$. Coal, \$2 perton of 2000 lbs.; evaporation, 8 lbs.

Allow 18×42 B to cost \$2500; but this requires \$400 for counter-shaft, \$300 for foundation and erection, and occupies extra space valued at \$100 per year.

$F_a = \$2500 + \$400 + \$300 = \$3200$. Counter-shafts and foundations require an interest-charge, and are liable to depreciation, being worthless when the engine is worn out.

$C_a = \$1200 + \$100 = \$1300$.

By substitution in the general equation we have:

$F_a = \$7333 + \$3125 = \$10,458$.

The money value of special expedients, as, for example, superheating steam, must be carefully studied from both sides the account. With moderate superheating a net advantage may probably nearly always be anticipated. Mr. Isherwood found, as an illustration, that with a yacht-boiler the following results were obtainable: *

Superheat. Fahr. deg.	Water per I.H.P. per hr.; lbs.	Gain, per cent.
62	13.5	34.7
37	14.5	30.0
30	15.9	23.0
18	17.3	16.5
2	18.8	9.0
0	20.7	0.0

The steam-pressure was about nine atmospheres, as a maximum.

Cost-values and Durability, in general, are connected by the following mathematical relation: †

Let y = first cost of less durable article;

x = first cost of more durable article to be on an equality with y;

n = number of years' duration of x;

n' = number of years' duration of y;

i = rate of interest per annum; for example, 5 per cent = .05.

$$x = y\frac{(1+i)n - 1}{(1+i)n-(1+i)n-n'}.$$

Value of x at different rates of interest ($y = 1$).

Duration, Years.		x at Rates of Interest per cent per Annum.			
n'	n	3	4	5	6
2	10	4.4604	4.3029	4.1539	4.0142
5	20	3.2491	3.0541	2.8783	2.7231
10	50	3.0162	2.6485	2.3644	2.1416
20	100	2.1237	1.8031	1.5926	1.4487

* Kennedy on the Serpollet coal-boiler; London Eng'g, 1891.

† Molesworth, p. 611.

The computation of interest is a common problem in this connection. The common expressions for such cases are the following:

Let $P =$ principal;
$n =$ number of years, integral or fractional;
$R =$ amount of \$1 or other unit at given rate per cent;
$r =$ interest " " " " " " "
$I =$ interest; $M =$ amount; $D =$ discount; $W =$ present worth;
$A =$ annuity to continue for n years;
$V =$ present value of an annuity.

Simple interest, $I = Pnr$; $M = P(1 + nr)$;

$$W = \frac{P}{1+nr}; \quad D = P - W = \frac{Pnr}{1+n}.$$

Compound interest, $M = PR^n$.

Annuities (simple interest), $M = nA\left(1 + \frac{n-1}{2}r\right)$;

$$V = \frac{nA\left(1 + \frac{n-1}{2}r\right)}{1 + nr}.$$

Annuities (compound interest), $M = \frac{R^n - 1}{R - 1}A$;

$$V = \frac{R^n - 1}{R^n(R - 1)}A.$$

238. Problems in Steam-boiler Construction are the following:*

(1) *The "Efficiency of the Steam-boiler"* is the ratio of the total quantity of heat utilized in the production of steam to

* Ibid., p. 475. The problems of steam-engine efficiency and finance are very fully treated in the concluding chapter of Part I of this work.

that set free in the combustion of the fuel. It has as the maximum limit unity, and is a function of area of heating-surface, and of factors dependent upon the character of the fuel and its combustion, and upon the design of the boiler.

(2) *The "Commercial Efficiency"* or the "*Efficiency of Capital*" employed in the maintenance of steam-generating apparatus of a *given power* is measured by the ratio of quantity of steam produced to the total cost of its continuous production. This efficiency is a maximum when that cost is a minimum.

(3) *The "Efficiency of a given Boiler-plant,"* or the commercial efficiency of a steam-boiler already in place and in operation, is a maximum when the work done by the boiler can be increased beyond that for which it was proportioned—if designed originally to give maximum efficiency of capital at a prearranged power, as above until the amount of steam made by that boiler *per dollar of working expense* is made a maximum.

These three efficiencies differ essentially in their character, and are determined by different processes.

In the first two cases, the variable element is usually the area of heating-surface per pound of fuel burned in the unit of time; in the last, the variable may be either the quantity of fuel burned or of steam made.

(4) *To what Capacity may any Given Boiler be forced without exceeding that Cost of Steam at which a Paying Profit is given?* is another problem in steam-boiler efficiency, which is of more frequent occurrence and usually more important than the preceding.

Taking the rate of conduction of heating-surfaces as varying as the square of the difference of temperatures of the gas and of the water on opposite sides of the sheet, the formula

$$E = \frac{1}{1 + \frac{ac'^2 W^2}{SH}}$$

is readily deduced, in which E is the efficiency, a a constant, c' the specific heat of the furnace gases, and W their weight; while H is the total heat expended and S the heating-surface. This expression is further transformed into

$$E_1 = \frac{BE}{1 + \frac{AF}{S}},$$

in which E is the theoretical evaporative power of fuel per pound, E_1 the probable actual evaporation in a boiler in which F is the weight of fuel burned on the unit of area of grate, and S is the area of heating-surface per unit of the same area.

A and B are here coefficients, having values, respectively, of 0.3 to 0.5 and 0.9 to 1 for bituminous coals, according to Rankine, and from 0.3 to 0.5 and from 0.8 to 0.9 with anthracite coal, as determined by experiments made by the Author. The lowest and best values of A are obtained when using a minimum needed air-supply; and the value of that coefficient is seen, by comparing the two equations just given, to vary as the square of the quantity of air supplied to the fuel. The value of B is dependent upon the character of the boiler, being greater as the design and construction are improved.

The following are illustrations of the results thus obtained:

Efficiency of Steam-boilers.

	I.	II.	III.	IV.
$\frac{F}{S}$	$A = 0.5;\ B = 1.$	$A = 0.3;\ B = 1.$	$A = 0.5;\ B = 1.$	$A = 0.3;\ B = 1.$
0.17	0.92	0.95	0.83	0.86
0.33	0.87	0.91	0.78	0.82
0.40	0.83	0.89	0.75	0.80
0.50	0.80	0.87	0.72	0.78
0.67	0.75	0.83	0.68	0.75

The expenses of operating a steam-boiler may be classed under three heads:

(1) Those costs of boiler and its maintenance which are

dependent upon the size and the character of the boiler itself and its attachments.

(2) Those costs of operation which are dependent upon the quantity of steam made and of fuel consumed.

(3) In addition to these variable expenses are often, perhaps usually, to be counted certain constant expenses which are unaffected by any change of proportions of boiler likely to be made in the assumed case.

A given amount of steam being demanded, it may be obtained either from a boiler so small as to use fuel extravagantly, or from a large boiler using fuel economically.

It is sometimes good finance to substitute large for small boilers. Thus one boiler 8 feet diameter by 30 feet long, pressure 100 pounds, being put in place of two boilers 7 feet diameter by 27 feet long, pressure 75 pounds, gave the following results:

With the two boilers 7 feet diameter were burnt 28 tons of coal a week; with the single boiler 8 feet diameter and a coal of rather better quality, 22 tons per week,—effecting a saving of about 10 per cent in money. The one boiler develops 350 I. H. P. It by no means follows that this may always be found the wisest course, risks as well as expenditures being taken into account.

239. The Designer's Problem seeks to determine what size and proportions are, on the whole, economically and financially best to develop a given power or produce a stated weight of steam. In each case arising in practice there will be found a certain proportion of heating-surface to grate-surface, and a definite size of boiler which will, on the whole, supply the desired quantity of steam most economically. Thus:

Let the total cost of fuel per annum and per pound burned per hour on the square foot of grate or on the square metre be called C. Let the total cost per annum of boiler, per square foot or per square metre of heating-surface be called D, and let $\frac{C}{D} = R$. In the first item is included Class 1, and in the second Class 2.

Then the cost of boiler maintenance per annum is DSG, where S is the area of heating-surface per unit of area of grate and G is the area of grate. The cost of fuel, etc., per annum, as per Class 2, is CFG, if F is the weight of fuel burned per unit of area of grate.

The total of costs variable with change of proportion of boiler is

$$P = DSG + CFG.$$

The profitable work of the boiler is measured by the quantity, by weight, of steam made, $FGE_1 = W$; E_1 being the evaporation of water per unit of weight of fuel.

The ratio of cost to work done is

$$y = \frac{P}{W} = \frac{DGS + CFG}{FGE_1} = \frac{CF + DS}{E_1 F}.$$

This quantity being made a minimum by variation of the area S, the most economical boiler is obtained.

But E_1 is a function of S, and, taking the value of E_1 from the equation

$$E_1 = \frac{BE}{1 + \frac{AF}{S}},$$

we obtain

$$y = \frac{(DGS + CFG)\left(1 + \frac{AF}{S}\right)}{BEFG} = \frac{DGS + ADFG + CFG + \frac{ACF^2G}{S}}{BEFG}$$

$$= \frac{DS + ADF + CF + \frac{ACF^2}{S}}{BEF},$$

which is a minimum when

$$S_1 = F\sqrt{\frac{AC}{D}} = F\sqrt{AR};\ \frac{S_1}{F} = \sqrt{AR}.$$

In illustration: Let a boiler, set in place, complete with all its appurtenances and in running order, cost \$3 per square foot of heating-surface, and the annual charges on all accounts entered in Class 1, above, be 20 per cent on this cost, the annual charge becomes $DS = \$0.60 \times S$ per square foot of grate, i.e., $D = \$.0.60$. Let the cost of operation, as for Class 2, amount to \$15 per annum per pound of fuel burned per hour on the square foot of grate; then $CF = \$15 \times F$; $C = \$15$; $\frac{C}{D} = R = 25$.

Assume $F = 10$ pounds of fuel per hour per square foot of grate, $A = 0.5$.

For this case, then, the boiler should have per square foot of grate,

$$S_1 = F\sqrt{AR} = 10 \times (0.5 \times 25)^{\frac{1}{2}} = 35;$$

35 square feet of heating-surface.

Similarly we get the following values:

COMMERCIAL EFFICIENCY OF BOILERS.

RATIO OF AREAS OF HEATING AND GRATE SURFACES.

Values of S.

F / *R*	6	10	12	15	20	30	40	50
25	21	35	42	52	70	105	140	175
16	17	28	34	42	56	84	112	140
9	12	21	24	32	42	63	84	105
4	8	14	16	21	28	42	56	70

The following are examples in greater detail of the application of the above:

EXPENSE ON BOILER ACCOUNT AND MAXIMUM COMMERCIAL EFFICIENCY.

CASES.	STATIONARY.		MARINE.	
	I.	II.	III.	IV.
Class 1 (*D*).	Cornish.	Tubular.	Tubular.	Tubular.
Total annual cost of boiler per unit of *S*.....	\$1.50	\$2.00	\$3.00	\$2.00
Interest..................................	.09	.12	.15	.12
Repairs and depreciation...................	.15	.20	.45	.30
Rent, insurance, and miscellaneous.........	.10	.07	1.00	.20
Total value of *D*..................	.34	.38	1.60	.62

Class 2 (C).				
Fuel (@ \$5 for I., II., IV.; \$4 for III.) per unit of F.	7.50	7.20	12.00	2.00
Transportation and storage	1.00	1.00	10.00	1.00
Attendance (variable cost)	0.00	0.50	0.50	0.00
Total	8.50	9.00	22.50	3.00
Value of $\frac{C}{D} = R$	25	23	14	5
Value of A	0.5	0.3	0.3	0.5
Value of $\sqrt{AR}$	3.5	2.7	2.0	1.6
Value of F	8	10	16	20
Value of $\sqrt{AR} = S_1$	28	27	32	32

240. **Proprietor's Problems.**—The owner's problems are of quite a different nature, thus:

A steam-boiler is in place and in operation; all constant expenses are known and all variable costs of maintenance and operation are determinable. The question arises: How much work can be obtained from the apparatus when driven to such an extent as to yield the maximum amount of steam per dollar of total cost of operation? The independent variable is now the quantity of fuel burned in the boiler, and this is, in the established equation, represented by F, the fuel burned per unit of area of grate. This problem is thus stated:

Given: All expenses, constant and variable, the method of variation of the latter, and the proportions of the boiler being given, to determine that rate of combustion which will make the commercial efficiency of the given plant a maximum.

For this case let K represent that total annual expense of working which is independent of Classes 1 and 2, and which falls into Class 3, and let $k = \frac{K}{G}$.

Let all other symbols stand as before.

Then the total cost of maintenance and operation will be

$$P' = kG + DGS + CFG,$$

while the work done will be, as before,

$$W = FGE_1.$$

The quantity to be made a minimum is, for the present case, the quotient of P' by W,

$$y = \frac{P'}{W} = \frac{k + DS + CF}{E_1 F},$$

F being taken as the independent variable.

This becomes a minimum when we substitute for E_1 its value $E_1 = \dfrac{BE}{1 + \frac{AF}{S}}$, and make the first derivative equal zero.

Then we find

$$F_1 = \sqrt{\frac{ks + DS^2}{AC}}.$$

When, in this expression for the value of F, giving maximum weight of steam for the dollar expended, we make $k = 0$, the expression may be reduced, as obviously should be possible, to the form shown already to be that giving the solution of the first problem:

$$S_1 = F\sqrt{AR}.$$

The following cases actually illustrate this problem:

EXPENSES OF BOILER AND MAXIMUM ECONOMY OF PLANT.

Cases.	Stationary. I.	Stationary. II.	Marine. III.	Marine. IV.
Cost of maintenance: D	\$0.34	\$0.58	\$0.88	\$0.62
Cost of operation: C	8.20	9.00	14.50	3.00
Cost of operation: K	30.00	25.00	10.50	10.00
For maximum fuel and work: F_1	16	13	17	21
For maximum efficiency, as before: F	8	10	16	20

Case No. 1 is that of a Cornish boiler, No. 2 that of a multitubular stationary boiler, No. 3 that of a sea-going steamer, and No. 4 that of a yacht.

It is seen that in all cases the weight of steam delivered from the boiler and the quantity of fuel burned at maximum commercial efficiency, for the case assumed, are less than where the boiler—once set and still capable of being forced to deliver

more steam than originally proposed and calculated upon—is worked up to a maximum delivery per dollar of total expense.

The similar problems for the steam-engine have been elsewhere treated.*

241. Problems in Lubrication of engines, machinery, and mills of various kinds, of a similar nature but of less definite form, often present themselves to the engineer and to the proprietor. The method of treatment of these problems adopted by the Author is as follows :†—

The total expense to be charged to the friction of machinery consists of the following items as principals, and probably of minor and less easily determined expenses : ‡

(1) The cost of power produced, only to be wasted by that friction.

(2) The expenses incurred in wear and tear of running parts, and in the replacement of parts destroyed, either by direct strains or by gradual wear due to such exceptional resistances as are the effect of excessive friction.

(3) The casual, the indirect, and often unperceived, yet none the less serious, losses throughout the system which are not included in the above.

(4) The cost of the lubricating materials applied for the purpose of ameliorating these losses.

The first item includes a part of the expense of the prime mover, such as cost of fuel and oil used on the motor, interest on the capital invested in the machine and in the machinery of transmission, wages paid engineer and fireman or other attendants, insurance and taxes upon that part of the plant, including so much of the building as is properly chargeable to

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* See Part. I. Also Jour. Franklin Inst., 1882; trans. Am. Soc. Mech. Engrs., 1882.

† Friction and Lubrication. New York, 1879.

On the Real Value of Lubricants, etc., Jan. 7th, 1885; Trans. A. S. C. E., vol. XIII. p. 476.

Friction and Lost Work in Machinery and Mill-work ; New York, J. Wiley & Son, 1885 ; chap. VIII.

‡ Trans. Am. Soc. Mech. Engrs., 1884.

the motive-power department. The second item includes costs of repair, refitting, or replacement of journals and of bearings; the repair of break-downs caused by excessive friction, or by hot bearings seizing the journals; and, often, the cost of throwing out the whole machine and introducing a new one to take its place. The conditions determining the life of the machine, in fact, are what are included under this head. The third item includes the exceptional damages resulting from friction of excessive amount, and which may be more likely to occur with one oil than with another. Its amount can never be calculated with any great degree of exactness.

Before the real value of any lubricant to the consumer can be determined, and whether any proposed change is desirable on the score of economy, it becomes necessary to ascertain the total expense chargeable to friction, in the manner already indicated, and to compare the difference of cost of unguents with the difference in costs of other items of expense produced by change of lubricant in the manner intended. In making this comparison it is first necessary to determine in what terms these expenses may be best expressed, and in what magnitude they enter the equations representing the problem.

(1) The cost of power wasted may be expressed in the usually adopted terms, the cost in dollars per horse-power and per hour, or per year; or it may be given in foot-pounds of work, irrespective of time. The first of these methods of valuation is the more common.

(2) The cost of wear and tear, and of depreciation, is an even more variable quantity than that of power. It cannot be stated definitely and generally; but it may usually be very fairly measured for any given case. An allowance is customarily made based upon the value of all machinery subject to this kind of depreciation. It will always be permissible to take this expenditure, in any establishment considered, as proportional to the power employed, and to include it thus in the first item. It sometimes happens that a decrease of the total power wasted by friction is accompanied by an increase in the

amount of wear: in such cases the oil producing this remarkable effect should be rejected.

(3) The third item, the casual and irregular losses, should, where possible, be made a constant and regular item of charge by securing insurance against all such kinds of loss. Where this cannot be done, the proprietor should insure himself by accumulating a fund to meet this expense, assuming a rate of accumulation which experience may determine to be safe for a series of years. This item then becomes chargeable as so much insurance, and can be introduced, with other insurances, in the first of the above divisions.

All three of the charges above described may be thus brought to one method of charging, and may be entered upon the account as so much per horse-power and per hour, or per year, or per foot-pound of work, if preferred.

(4) The fourth and last item, the cost of lubricant, is measured by the charge per gallon, and by the number of gallons used per hour or per annum, or for the specified work. This is ascertainable by observation or trial, either in the establishment in which it may be used, or upon the testing-machine, under the conditions, if attainable, as to pressure, speed, temperature, and condition of bearing surfaces, of its application in the shop, the mill, or on the axle of a railway-train.

The algebraic theory of this case is constructed as follows:

If in any case U be taken as the measure of the power wasted or of the work lost per hour or per annum, and if k' be its total cost, as above, for the unit of time, and if q be the quantity of the lubricant used in the same time and k is its cost, including its application to the journals, the total expense chargeable to friction, being called K, must be measured by

$$K = kq + k'U. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

But the work done is equal to the product of a constant, a, dependent upon the units employed, the value of the coefficient of friction f, the total load P, and the space passed over by the rubbing surfaces S, which last is also equal to the product

of the velocity of rubbing V, and the time t, taken as the unit of comparison.

Then

$$U = afPS = afPVt, \quad \ldots \ldots \quad (2)$$

and the cost K thus becomes

$$K = kq + k'afPS, \quad \ldots \ldots \quad (3)$$

or

$$K = kq + bf; \quad \ldots \ldots \quad (4)$$

in which b is a constant in any given case and equal to

$$b = ak'PS, \quad \ldots \ldots \quad (5)$$

and the equation, as thus simplified, may be applied to all cases.

The value of f is to be ascertained by experiment, under the exact conditions of use; and this being determined, the total cost becomes calculable, and a satisfactory comparison may be made. That lubricant is best which, all things thus considered, gives the lowest value of K, and where wear of serious amount occurs, as may happen when using oils badly adapted to the pressure and speed, the cost of such additional wear must be put in as an additional charge.

The value of the oil, in terms of the total cost and of cost of wasted power, is

$$k = \frac{K - bf}{q}; \quad \ldots \ldots \quad (6)$$

which equation shows that the value is the greater as the quantity demanded is less, and that it also increases when the coefficient of friction decreases. These equations also show that the total cost is very nearly proportional to the coefficient of friction when, as is the usual case, the cost of lubrication is small in comparison with that of the power wasted. The value of the lubricant is then very nearly proportional to the reciprocal of the coefficient of friction, and has no necessary or direct relation to the market-price.

Two oils being compared, the costs of wastes are, respectively,

$$K_1 = k_1 q_1 + bf_1 ; \qquad K_2 = k_2 q_2 + bf_2 . \quad . \quad . \quad . \quad (7)$$

and the saving effected by the use of the better lubricant is

$$K_1 - K_2 = k_1 q_1 - k_2 q_2 + b(f_1 - f_2) ; \quad . \quad . \quad . \quad . \quad (8)$$

If the saving in value of power lost is just equal to the difference in cost of lubrication, it is evident that the change will not affect the total cost, and it is a matter of indifference whether it is made or not. That is to say, if $b(f_2 - f_1) = k_1 q_1 - k_2 q_2$, we shall have $K_1 = K_2$, and profit and loss are equal. When $b(f_2 - f_1) > k_1 q_1 - k_2 q_2$, the result is evidently a gain; and when the first member of the inequality is less than the second, a loss is effected. Thus we have a criterion of the advisability of changing the lubricant in the equations

$$K_1 = K_2 ; \qquad k_2 q_2 - k_1 q_1 = b(f_1 - f_2) ; \quad . \quad . \quad . \quad (9)$$

$$k_2 = \frac{k_1 q_1 - b(f_2 - f_1)}{q_2} . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If two oils are compared, therefore, it is seen that, the first having the price, k_1, giving the coefficient, f_1, when used in the quantity, q_1, the second may be profitably used in the quantity, q_2, if giving the coefficient, f_2, only when it can be purchased below the price, k_2, of equation (10), the two prices being considered as including the cost of application to the bearing and of removal. Should the oils compared have so little body that wear takes place to any appreciable extent, the cost of the wear is to be added to the cost of power in each case.

If, in any case, as often happens, the quantity used is practically the same, whichever oil is used, $q_1 = q_2$, and the criterion becomes:

$$k_2 - k_1 = \frac{b}{q_2}(f_1 - f) ; \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$k_2 = k_1 - \frac{b}{q_2}(f_2 - f_1) . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The allowable purchasing price is below the value thus obtained.

Where the same oil is used, but may be applied in greater or less quantity, we may obtain, similarly, a criterion for the quantity to be profitably used. It is evident that the advantage of increasing the quantity is to be found in the reduction of the cost of power and incidental losses. If, in any two cases, we get

$$k(q_2 - q_1) = b(f_1 - f_2) \quad . \quad . \quad . \quad . \quad . \quad (13)$$

the gain just balances the loss, and the criterion becomes

$$q_2 = q_1 + \frac{b}{k}(f_1 - f_2) \quad . \quad . \quad . \quad . \quad . \quad (14)$$

and, assuming it to be found, as is usual, that a decrease of power follows increase of freedom of supply of lubricant beyond the amount customarily given, the limit is reached at the above amount.

This last statement must, however, be qualified by the reminder that it is often possible to supply oil as freely as may be desired, without important loss, by the use of a good system of collection and renewal, with occasional purification. In such case, the cost of oil actually condemned and rejected is that to be ascertained and introduced into the equation. The comparison then lies as in the first cases, the costs including those of purification and replacement.

As the friction of lubricated surfaces is sometimes affected to a very great extent by variation in the rate of supply, especially at high speeds of rubbing, this case becomes important. The lower the cost of the oil, in any case, and the higher the cost of power, the more freely should the unguent be supplied to the bearing.

When the relative durability and the coefficients of friction of oils proposed in any case are known by direct experiments made under the precise conditions of intended use, it is equally easy to determine, wear being neglected, what are their relative money values. If an oil already in use be taken as the

standard, and if an oil to be compared with it is found to have e times the endurance of that standard, the quantity used should be $q_2 = \frac{q_1}{e}$. If the second oil have a coefficient of friction h times as great as the first, the work of friction will be correspondingly reduced or increased, and the cost of that work will be $bhf_1 = bf_2$. The total costs will then become

$$K_1 = k_1 q_1 + bf_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$K_2 = k_2 q_2 + bhf_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

and the criterion is obtained, as before, by making these expressions equal, whence

$$K_1 = K_2; \quad k_2 = \frac{[k_1 q_1 + bf_1(1 - h)]\,e}{q_1} = ek_1 + bef_1 \frac{1 - h}{q_1}. \quad (17)$$

Any cost, on the journal, less than this value of k_2 is profitable; any higher cost will produce a loss. At this cost the user will neither gain nor lose by the change of lubricant.

It is obvious that the value of b may be expressed in any convenient units of cost quite as well as in those above taken. It is usual to measure costs on railways by amounts per train-mile, as, for example, pounds per train-mile, in measuring expenditures of fuel, miles run per quart, or per gallon, in rating expenditures of oil. As has been seen, these are but a part of the total expense; but it may sometimes be convenient to obtain an approximate estimate of the relative values of lubricants, on the assumption that wear and other costs may be neglected, and the value of b is then to be expressed in the money value of fuel used per train-mile; which cost will evidently be proportional to f. The relative costs will be proportional to the values of h.

In railway practice it is often found that the cost of wear is a very serious item, but probably usually when using an oll which is unfitted for use as a lubricant, and which should never be used at all. Could the lubrication be as effectively carried on as in other cases, and could the dust be perfectly excluded,

the loss from this cause would become insignificant. Assuming this to be the case, we may write the equation for this case

$$K = kq + df, \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

in which df is the total cost of power per train-mile.

The criterion is found as before, substituting d for b, in the several equations already derived,

$$k_2 = \frac{k_1 q_1 - d(f_2 - f_1)}{q_2}. \quad . \quad . \quad . \quad . \quad . \quad (19)$$

As, in all cases, the cost of power wasted is a function of the quantity q, if the law of variation can be ascertained and expressed algebraically, the most economical rate of supply can be ascertained in making $\frac{dK}{dq} = 0$, for a minimum. Similarly, the best value of k may be determined, for the case in which the same quantity is always used, by making $\frac{dK}{dk} = 0$.*

The figures given below will be taken here as fair prices of the unguents named, as sold in the market. In each case the oil is assumed to be a good representative of its grade, such as may reasonably be expected to be obtainable in the market by any skilful and experienced buyer who may choose to secure it.

PRICES OF LUBRICANTS.

Sperm oil	per gallon,	$1 10
Neats-foot	" "	1 00
Tallow oil	" "	0 70
Lard oil	" "	0 70
Greases	per lb.,	0 25
Heaviest mineral oil	per gallon,	0 75
Medium machinery-oil	" "	0 50
Light lubricating oil	" "	0 25
Crude well oils	" "	0 20
Kerosene (unrefined)	" "	0 10

* Encyclopædia Britannica, art. "Lubricant."

These figures can only be taken as illustrative. The prices obtained in the market for the machinery oils vary enormously, and without any fixed relation to their values. In some cases the buyer is controlled in his selection by other circumstances than their friction-reducing power and their price—as where "cylinder-oils" for engines are called for, when the high fire-test mineral oils only can be used.

It may probably be estimated that at least one half of all the power expended in the operation of the average manufacturing establishment is applied to the work of overcoming the friction of lubricated surfaces. The coefficient of friction will average a high or a low figure according to the kind of machinery. The heavier the latter, the lower the friction coefficient. Light machinery gives a high value of friction, which is therefore very great on the spindles, and on the machinery generally, of the textile manufactures, lower on the heavy machines of the iron-working trades, and very low, comparatively, on the axles of engines and railway cars. It will be here taken as averaging, in good practice, ten per cent in mills, five per cent on heavy machinery, and one per cent on railways. The oil must evidently be selected with a view to its use, the heavier pressures and lower coefficients being necessarily obtained with heavy lubricants—oils or greases—and the lower pressures and higher coefficients being given where the lightest possible oils are properly and customarily employed with more copious supply.

Mr. Edward Atkinson finds that, in fifty-five mills, working on similar fabrics, and among which a variation of 20 per cent should not have been expected, the actual range of cost of lubrication was 350 per cent. Subsequently, careful management reduced the average from $10.03 to $6.67 per 10,000 pounds of cloth made. The oils found in use, and tested, varied in quality to the extent of 300 per cent. The best oils for these mills were reported by Mr. Woodbury as being mineral oils mixed with some sperm or lard, and having a gravity of about 28 to 32 Beaumé (S. G. 0.886 to 0.864).*

* Transactions of the American Soc. Mech. Engineers, vol. IV., p. 319.

The total power used in these mills is found by Mr. Henthorn to average about 0.75 horse-power per loom, or 15.75 horse-power per 1,000 spindles, and to vary from 0.5 to nearly one horse-power per loom, or from 11 to 22 horse-power per 1,000 spindles. The same authority finds the power demanded by engine and shafting alone to form from 17 to 34 per cent of the whole. The smaller of these figures represents the best practice, the higher figures show what may be expected with faulty arrangement and bad lubrication. The quantity of cloth made in New England mills ranges from about 2,000 pounds per annum, per horse-power, up to above 3,000. A variation of temperature, such as occurs between winter and summer, causes a variation of ten per cent in the production of cloth, the greater amount being obtained in summer. A mill making print-cloths, four threads to the inch, and of No. 32 yarn, with frame-warp and mule-filling, produces about one pound of cloth per spindle per week, and demands about 16 horse-power per 1,000 spindles. Thus 10,000 pounds of cloth per annum requires 200 spindles, and proportional plant costing about, at present prices, or a little above, $11 per spindle for all machinery and buildings, exclusive of stock, land, and live capital, and the power demanded is not far from 3.3 horse-power. One horse-power may drive as few as 150 or as high as 300 spindles, at 10,000 revolutions, with bad or with good oils, a fair average being, for empty cotton-spindles, not far from 250.

The quantity of oil used in cotton-mills varies from 10 to 30 gallons per 10,000 pounds of cloth made per annum, averaging not far from two gallons per horse-power per annum, and costing from 20 cents to $1 per gallon, averaging probably not far from 50 cents.

A machine-shop uses, properly, a much heavier oil than a cotton-mill, and much less in proportion to power employed. One of the largest and best-known steam-engine-building establishments in the country uses 120 horse-power, and consumes 350 gallons of sperm oil for the heaviest machinery, such as planers, and for the wood-working tools; 100 gallons

of finest heavy mineral engine-oil for the engine; 300 gallons of mineral or mixed oil, at about 40 cents per gallon, for shafting and tools; and expends a total, on lubrication, of about $600 per year, an average of about 80 cents per gallon. Another and smaller shop, also well known to the Author, employs but 30 horse-power, and uses 175 gallons of oil per year for lubrication, the average cost being but 18 cents per gallon. Still another establishment, building tools and light machinery, uses an estimated amount of 100 horse-power, and 700 gallons of oil at 20 cents per gallon, and for grease used on the main line of shafting, 25 cents per pound. A similar shop of somewhat larger size reports an expenditure of but 40 gallons per year on 550 bearings, the oil being supplied by hand, a certain number of drops at a time, at regular intervals. Probably a fair estimate for the heavy class of machinery found in such establishments may be about 0.0002 gallon per horse-power per hour.

A large pumping-engine using about 0.00015 gallon per horse-power per hour exhibits a low consumption for that class of machinery.

Railway work is probably more exacting in its demands, and more variable in its practice in regard to quantity of oil used, than any other class of machinery consuming lubricants. The consumption of oil is usually reckoned per train-mile, and the following are the figures given by one of the best of the Massachusetts roads for one month on the engines alone:

	Coal—Lbs. per Mile.	Oil—Gallons.
Best express engines	43	0.009
Best freight engines	55	0.0094
Average passenger engines	50	0.009
Average freight engines	61	0.0084

The average of all Massachusetts roads in one year was: for wages on the engine $0.28; for fuel, $0.115; and for oil and waste, $0.0105. On the Pennsylvania Railway, between New

York and Pittsburgh, in 1883, the totals were reported as follows:

Total mileage	15,625,478
Coal, tons of 2000 pounds	743,020
Wood, cords	13,685
Oil, quarts	613,478
Tallow, pounds	721,992
Waste, pounds	267,158
Tons moved one mile	2,996,893

The coal averaged in price but $1.00 per ton, exclusive of freight charges to the road, the wood cost $2.88 per cord, the oil 28 cents per gallon, the tallow and waste each 8 cents per pound.

The cost of steam-power, which is usually the principal item included in the cost of the wasted work of friction, varies greatly with size and kind of engine, character of fuel, expenses of operation, and with all the items such as insurance, repairs, and depreciation, incidental to its use. A fair figure is, perhaps, for ordinary mill engines of moderate power, $0.02 per horse-power per hour, and double this amount is not unusual. Of this total, from 50 to 80 per cent may be assumed to be, on the average, the cost of fuel. Adding the incidental costs, it may be considered a fair estimate for such cases to take the total charge at $0.03 per horse-power and per hour.

The actual cost of steam-power in mills ranges from as low as $50 to as high as $100 per horse-power, according to circumstances. The latter figure represents the cost of the best modern machinery. The interest on first cost may be assumed at 6 per cent, the appropriation for a sinking-fund at 2½ per cent, the working expenses at not far from 10 per cent in good cases, and the cost for fuel, on the average, at about 20 per cent of the cost of the plant. The total cost thus calculated will vary from, perhaps, $35 to nearly or quite $100 per annum per horse-power. It has been taken above at $60, and about 50 per cent of its own amount added for miscellaneous costs not included in the direct calculation.

In illustration of the application of these principles, the following cases, which are examples of practice falling within the experience of the Author, may be given:*

(1) In a machine-shop using about 100 horse-power of which one-half is supposed to be applied to the overcoming of the friction of lubricated surfaces of journals and their bearings, it is found that the cost of power is very nearly $100 per horse-power per annum, inclusive of all of the incidentals above mentioned. The average coefficient of friction is not far from 0.05; the oil used costs, on the average, $0.50 per gallon, consisting mainly of lard and heavy mineral oils, and is supplied at the rate of 0.02 gallon per working hour, the working year consisting of 3000 hours.

Then, if 50 horse-power should be used in the work of overcoming frictional resistances, the cost of power would be $5000 per annum, or $1.67 per hour, which is represented in the equations already given by bf_1. Since f_1 is found to be 0.05, b is equal to 33.333. The oil used being found to cost, in place on the journal, $0.50 per gallon, and to be used at the rate of 0.02 gallon per hour, the total cost of lubrication is $0.01 per hour. Hence we have (Eq. 4):

$$K_1 = k_1 q_1 + bf_1 = 0.01 + 1.67 = \$1.68 \quad . \quad . \quad . \quad (A)$$

or $5040 per annum.

Should it be proposed to make a change of oil, using oil costing but $0.25 per gallon, and of which 0.03 gallon per hour will be demanded, and which will make the coefficient of friction 0.06, the cost of power will be increased one-fifth, and that of oil diminished one-fourth; the equation then reads:

$$K_2 = 0.25 \times 0.03 + 33.333 \times 0.06 = \$2.0075 \quad . \quad . \quad (B)$$

equal to $6022 per annum.

The gain effected in cost of oil is one fourth of one cent per hour, while the loss in cost of wasted power is 33.333

* Friction and Lost Work.

cents per hour. In other words, a gain of \$7.50 per annum on the books of the purchasing agent or proprietor is to be charged against a loss, in the cost of running the establishment, of \$1000. The net loss is \$982.50 per annum.

Should it prove possible to adopt a system of oil-bath, or other method of free lubrication, so as to bring down the coefficient to 0.02, as is not at all unlikely to prove practicable, and assuming that four times as much oil, of the second quality, is used as in the last case, we shall have

$$K_2' = 0.03 + 0.666 = \$0.696 \text{ per hour}, \quad . \quad . \quad \text{(C)}$$

\$2088 per year, producing a gain of two thirds the total cost of lost work, as in the last case. This amounts to nearly \$4000 per year, or, as compared with the present running expense, as given in the first case, to nearly \$3000. The annual cost of oil in the three cases amounts to \$30, \$22.50, and \$90, respectively; and it is at once seen that in this example of application the saving, actual or possible, to be effected by any bargain made in the oil market is insignificant in comparison with that to be produced in the shop by careful lubrication. A system of collection and purification of the oil running off the journals into the drip-pans may in nearly all cases be easily adopted, which will at once reduce the cost of lubricant, and make its first cost a matter of still less consequence.

Finally, suppose a grease used in this shop, such as now costs 25 cents per pound, and assume that it is given as a sample, costing the proprietor nothing, but bringing up the coefficient of friction, as an average, to 0.10. The cost of power is now the total expense, and this becomes

$$\$3.333 \text{ per hour}, \quad . \quad . \quad . \quad . \quad . \quad \text{(D)}$$

or \$10,000 per annum: while the loss to the owners of the establishment, on their bargain, is \$5000 per annum.

It will next be asked: What price represents the limit which may not be exceeded, without loss, in the purchase of the oils proposed to be substituted for that first used in this

instance? This question is answered by the application of the criterion established by equations (10) and (14). Thus, comparing cases (A) and (B), we have

$$k_2 = \frac{k_1 q_1 - b(f_2 - f_1)}{q_2} = \frac{1.01 - 0.333}{0.03} = -\$10.78.$$

The second oil causes a loss of \$10.78 for every gallon used, and hence cannot be used without loss, unless the user is paid that sum to take it and apply it to his machinery.

Comparing cases (A) and (C), using equation (10),

$$k_2 = \frac{0.01 + 1.00}{1.2} = \$0.84\,;$$

and it is found that the second disposition of the poorer grade of oil is of such advantage that it is as well worth \$0.84 per gallon as is the better oil worth \$0.50, used as at first proposed, and as is customary. But it would be a still better investment, in all probability, to purchase the better oil, and to use as in the case compared.

Comparing cases (A) and (D), using equal amounts per gallon,

$$k_2 = \frac{00.0 - 33.333 \times 0.05}{0.02} = -\$83.44\,;$$

and the heavier lubricant is found to subject the user to an expense amounting to over \$10 for each pound used. It must not, however, be from this inferred that it is always wasteful to use the greases. They are often advantageous where exceptional pressures are used or troublesome bearings are met with, and are sometimes absolutely indispensable, saving large amounts by their reduction of expenses in the cooling and preservation of journals and renewal of bearings. In the above case it is probable that a much smaller quantity of grease than of oil would have sufficed, which would have reduced the total cost of grease, if purchased, but would have proportionally

increased the loss to the proprietor, both absolutely and as reckoned per pound of unguent applied.

(2) As a second illustration, assume a cotton-mill to use a good oil, averaging \$0.70 per gallon, at the rate of 0.7 gallon per hour, with a mean coefficient of friction 0.10, on machinery demanding 400 horse-power, of which 120 horse-power is required to overcome the friction of surfaces lubricated by the oil. Taking the value of the power at \$65 per horse-power per annum, and 3000 working hours, we have $b = \$26$. If it is proposed to substitute for the oils used in this mill others averaging a cost of \$0.40 per gallon, giving a mean coefficient of friction of $f = 0.12$, and of which one gallon will be used per hour, we shall have

$$K_1 = 0.49 + 2.60 = \$3.09,$$
$$K_2 = 0.40 + 3.12 = \$3.52,$$

and a gain of 9 cents per hour, or \$270 per annum, in buying oil, is to be set against a loss of 52 cents per hour, or \$1560 per year, in increased expenses on the account of operating the mill, the net loss amounting to above one thousand dollars per year.

Applying the criterion to this case, we have

$$k_2 = \frac{0.49 - 0.52}{1.0} = - \$0.03,$$

as the loss on each gallon of the second lubricant. The owner of the mill cannot afford to accept it, in substitution for the better oil, as a gift. The substitution of an engine-oil on the spindles for the best spindle-oil might readily double the expenditure of power absorbed by the spinning machinery, and thus increase the cost of both lubrication and power, the former having both a higher coefficient of friction and greater price than the latter.

(3) In further illustration, assume a railway-train to be supplied with a good standard lubricating-oil for engine and

axles, costing on the journal \$0.25 per gallon, and to use 0.02 gallon per train-mile, the coefficient of friction, when everything is in good order and all journals cool, being 0.01. Taking as a fair figure \$0.10 per mile for costs of power and incidentals variable with power, and presuming that, under the circumstances, wear may be reduced to an unimportant amount, and may be neglected, the relative costs of lubricating material and of power may be introduced into equations (18) and (19), as in the above examples. We thus obtain

$$K_1 = k_1 q_1 + df_1 = 0.005 + 0.10 = \$0.10\tfrac{1}{2},$$

as the total money loss due to the existence of friction.

If it be proposed to substitute for the oil in use a cheaper oil, costing on the journal \$0.15, and of which fifty per cent more will be used, and which will give a coefficient of friction 0.015,—a not uncommon case,—the total cost becomes

$$K_2 = 0.15 \times 0.03 + 10 \times 0.015 = 0.15\tfrac{1}{2};$$

and it is found that a gain of one twentieth of a cent per mile, in cost of oil, is met by a loss of one hundred times as much, or five cents per mile, in cost of power. Should the second oil give increased wear, its cost must be added to the account of losses produced by the change. Had the second oil been used in the same quantity as the first, $1\frac{1}{2}$ cents per mile would have been saved over the last figures, and the loss would be then $3\frac{1}{2}$ cents per mile.

To determine what could be paid for the second oil, as used, in order that no loss should take place in consequence of the change, equation (17) is to be used, and this gives

$$k_2 = \frac{0.005 - 10 \times 0.0005}{0.03} = \$0.00;$$

that is to say, the real value of the oil to the consumer is just 0, if the oil at first used was worth 25 cents per gallon.

The conclusions to be drawn from the principles and the theory which have been presented in this paper, and from the examples of application to practice which have been introduced as fairly representing their use in various departments of engineering, are obvious and definite:

(1) To secure the highest possible efficiency of machinery, and maximum economy in the operation of establishments in which it is employed, lubricants must be very carefully selected with reference to the precise conditions, as to pressure, velocity of rubbing, etc., met with in the individual case.

Where—as in machine-shops and mills, for example—there exist great differences in these respects, it will be found advantageous to use different oils, as heavy oils on the engine-bearings, special "cylinder-oils" in the steam-cylinder, lighter oils on the shafting, and the lightest of the better classes of lubricating-oils on light machinery, as on spindles.

(2) Differences in price of oils, or other lubricants, are usually of exceedingly slight importance in comparison with differences in costs of power; and the value of the coefficient of friction is therefore of vastly greater consequence than either the price of the unguent or its endurance.

(3) The best oils for specified purposes should be taken, as a rule, whatever their market-price; while the oils which are not well adapted to the purpose in view canno tbe economically purchased at any price. Thus, also, tallow containing acid used in its purification, or liable to set free any of the fatty acids of which it is in part composed, may injure valve-seats, piston, rings, or the cylinder itself seriously, if employed as a cylinder-oil, or may corrode the boiler when passing with the feed-water. In such case its value is *negative* and important.

The actual costs of lubrication are enormously variable: but no estimate has any value which does not include the costs of work of friction. Cylinder oils, in ordinary mill-engines, cost from less than one cent per horse-power per hour upward, and the quantity used should not exceed one pint per day for a 500 horse-power engine. With cotton-spindles the number per horse-power, at 10,000 revolutions per minute, may vary,

with the character of the lubrication, from 150 to 300 or more. On large steam-vessels, 0.001 gallon per mile, for all lubrication, need not be exceeded with the best oils; but actual cost in practice varies from 0.0005 gallon to twenty times that figure per mile. One to two quarts per 100 miles are usual figures. The U. S. S. Kearsarge has steamed 3000 miles with a consumption, in engine-cylinders, of less than 1½ gallons of oil, and 1500 miles on one quart. One drop in two minutes is a rate of feed sometimes adopted.

Oils are often "tested" by subjecting them to a pressure similar to that under which they are to be employed, and working at a speed of rubbing also as nearly as practicable similar to that of proposed use, on a testing-machine designed for the purpose, as in that here illustrated, in which provision is made for observing the friction-resistance, and the varying temperature of the journal. The swing of the pendulum, *I*, over the arc, *PP'*, gives a measure of the latter, and the thermometer, *Q*, the heat of the bearing. The journal is clasped by the "brasses," with a pressure adjusted through the helical spring enclosed in the pendent arm at *M*.*

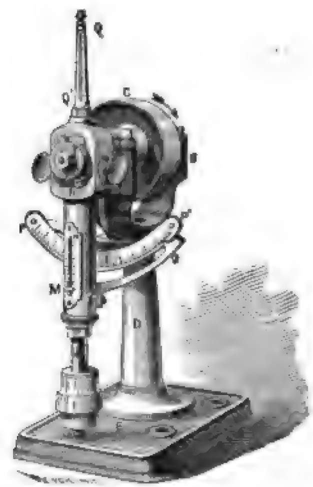

Fig. 215.—Thurston's Oil-testing Machine.

242. Estimating Costs, Expenditures, and Receipts thus constitutes an essential part of the work of the engineer, whether in designing machinery, operating that already constructed, or advising, as consulting-engineer, in regard to any matter in which, as in nearly every engineering question, finance enters as the final and vital consideration. In this work the engineer not only studies his own data and the fruits of his own experience, but, in every case of importance, calls

* Friction and Lost Work.

to his aid those who may have had special experience and who have acquired special knowledge by long practice in those departments in which he is himself less at home. In this work. if in any, the specialist is needed. The processes and principles which have been here described, and those which every practising-engineer is continually devising and learning in the course of his own work, have often enormous value.

Mr. Henthorn makes the costs of operation of mill-engines, simple and compound, as in the table on pp. 867, 868 :*—

Mr. Henthorn assumes the cost, at that date, of compounding engines at $4.00 per rated horse-power, and computes the annual saving by the operation as given in the next table, taking the probable average saving at 20 per cent, or one-half pound of fuel per I. H. P. per hour, the cost of coal at $4 50 per ton, and the running-time at 3090 hours per year.

Designing a "Steam-plant" is often a matter of as great complexity as of importance. It includes the consideration of the location, the best arrangement, and the most convenient and inexpensive method of combination of the whole system of boiler, engine, and machinery with reference to receipt and delivery of material, cost of buildings and mechanism of transmission, distribution of power, and convenience of operation and attendance.

The whole system—boilers, engine, and machinery of transmission—should be arranged as compactly, and all as near the machinery to be driven, as is practicable. The boilers, if of the older type of "shell boiler," should usually be placed in a separate boiler-house so located and arranged that an accident shall not seriously endanger adjacent property or human life. The engine and boilers should be placed in close juxtaposition, in order that the length of steam-pipes may be as small as possible, and so that the eye of the responsible director of both may most readily supervise boilers and engines and their attendants. Both engines and boilers should, if possible, be so located that the machinery of transmission may be as light,

* Report N. E. Cotton Mfrs' Assoc., 1887

COST OF SIMPLE AND COMPOUND ENGINES.

	Horse-Power of Engine.					
	250 H. P.	300 H. P.	400 H. P.	500 H. P.	600 H. P.	700 H. P.
Steam-jacketed Condensing Engines.	Single.	Single.	Single.	Single.	Single.	Pair.
Size of Engine	23″ × 60″	26″ × 60″	30″ × 60″	32″ × 60″	34″ × 72″	28″ × 60″
Diameter and Face of Main Driving-pulley	20 ft. × 31″	22 ft. × 33″	25 ft. × 39″	25 ft. × 52″	26 ft. × 60″	27 ft. × 66″
Approximate Cost of the Machinery complete with Condensing Apparatus, Boilers, Flues, Pumps, Heaters, Piping, etc.	$10,780 00	$12,830 00	$15,830 00	$18,480 00	$22,340 00	$28,010 00
Approximate Cost of the Foundations for Engines and Boilers	1,675 00	2,010 00	2,680 00	3,350 00	4,020 00	4,690 00
Total Cost of the Machinery and Foundations ready for Steam	12,455 00	14,840 00	18,510 00	21,830 00	26,360 00	32,700 00
Yearly Cost of supplying the above Engine with Fuel for 309 Days of 10 Hours each, and Coal at $4.75 per ton of 2240 lbs., on a Basis of 2.55 lbs. per Indicated Horse-power, and including all Fuel	4,177 18	5,012 62	6,683 49	8,354 37	10,025 24	11,696 11
Steam-jacketed Compound Engine. Steam, 125 lbs. Pressure.	Single.	Single.	Single.	Single.	Single.	Pair.
Size of Engine	13 & 26 × 60″	15 & 30 × 60″	15 & 30 × 72″	17 & 34 × 72″	18 & 36 × 72″	14 & 28 × 72″
Diameter and Face of Main Driving-pulley	20 ft. × 31″	22 ft. × 33″	25 ft. × 39″	25 ft. × 52″	26 ft. × 60″	27 ft. × 66″
Approximate Cost of the Machinery complete with Condensing Apparatus, Boilers, Flues, Pumps, Heaters, Piping, etc.	$12,380 00	$14,580 00	$19,400 00	$21,520 00	$25,254 00	$31,077 00
Approximate Cost of the Foundations for Engines and Boilers	2,050 00	2,460 00	3,280 00	4,100 00	4,920 00	5,740 00
Total Cost of the Machinery and Foundations ready for Steam	14,430 00	17,040 00	22,680 00	25,620 00	30,174 00	36,817 00
Yearly Cost of supplying the above Type of Engine with Fuel for 309 Days of 10 Hours each, and Coal at $4.75 per Ton of 2240 lbs., on the Basis of 1.7 lbs. per Hour, per Indicated Horse-power, and including all Fuel used	2,784 78	3,341 73	4,455 65	5,569 56	6,683 47	7,797 39
Difference in the Yearly Cost of Fuel-supply between the Two Types of Engines	1,392 40	1,670 89	2,227 84	2,784 81	3,341 77	3,898 72
Additional First Cost of the Machinery and Foundations for a Compound Engine over a Simple Steam-jacketed Condensing Engine of like Power	1,975 00	2,200 00	4,170 00	3,790 00	3,814 00	4,117 00

COST OF SIMPLE AND COMPOUND ENGINES.—*Continued.*

	Horse-power of Engine.				
	800 H. P.	900 H. P.	1,000 H. P.	1,150 H. P.	1,400 H. P.
Steam-jacketed Condensing Engines.	Pair.	Pair.	Pair.	Pair.	Pair.
Size of Engine	30" × 60"	30" × 60"	32" × 60"	32" × 72"	36" × 72"
Diameter and Face of Main Driving-pulley	27 ft. × 74"	28 ft. × 80"	30 ft. × 84"	30 ft. × 100"	30 ft. × 115"
Approximate Cost of the Machinery complete with Condensing Apparatus, Boilers, Flues, Pumps, Heaters, Piping, etc.	$31,110 00	$32,940 00	$37,010 00	$42,040 00	$50,268 00
Approximate Cost of the Foundations for Engines and Boilers	5,360 00	6,030 00	6,700 00	7,705 00	9,380 00
Total Cost of the Machinery and Foundations ready for Steam	36,470 00	38,970 00	43,710 00	49,745 00	59,648 00
Yearly Cost of supplying the above Engine with Fuel for 309 Days of 10 Hours each, and Coal at $4.75 per Ton of 2240 lbs., on a Basis of 2.55 lbs. per Indicated Horse-power, and including all Fuel	13,366 99	15,037 86	16,708 74	19,215 05	23,392 23
Steam-jacketed Compound Engine. Steam, 125 lbs. Pressure.	Pair.	Pair.	Pair.	Pair.	Pair.
Size of Engine	15 & 30 × 72"	16 & 32 × 72"	17 & 34 × 72"	18 & 36 × 72"	21 & 42 × 72"
Diameter and Face of Main Driving-pulley	27 ft. × 74"	28 ft. × 80"	30 ft. × 84"	30 ft. × 100"	30 ft. × 115"
Approximate Cost of the Machinery complete with Condensing Apparatus, Boilers, Flues, Pumps, Heaters, Piping, etc.	$34,150 00	$37,900 00	$42,190 00	$48,238 00	$57,300 00
Approximate Cost of the Foundations for Engines and Boilers	6,560 00	7,380 00	8,200 00	9,430 00	11,480 00
Total Cost of the Machinery and Foundations ready for Steam	40,710 00	45,280 00	50,390 00	57,668 00	68,780 00
Yearly Cost of supplying the above Type of Engine with Fuel for 309 Days of 19 Hours each, and Coal at $4.75 per Ton of 2240 lbs., on the Basis of 1.7 lbs. per Hour, per Indicated Horse-power, and including all Fuel used	8,911 30	10,025 21	11,139 13	12,809 90	15,594 78
Difference in the Yearly Cost of Fuel-supply between the Two Types of Engines	4,455 69	5,012 65	5,569 61	6,405 06	7,797 45
Additional First Cost of the Machinery and Foundations for a Compound Engine over a Simple Steam-jacketed Condensing Engine of like Power	4,240 00	6,310 00	6,680 00	7,923 00	9,132 00

ORDINARY DAILY AND YEARLY RUNNING EXPENSES.

Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Per Cent of Exhaust-steam used.	Lbs. Coal per I.H.P. per Hour.			Cost of Cool per I.H.P. per Day of 10½ Hours. @ $5.00 per Long Ton—2,240 Lbs.			Attendance of Boilers per I.H.P. per Day.			Attendance of Engine per I.H.P. per Day.			Oil, Waste, and Supplies per I.H.P. per Day.			Total Daily Expense.			Total Yearly Expense—308 Days.		
	Compound.	Condens'g.	High-press.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.
				c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	$	$	$
0	1.75	2.50 2.62	3.00	4.00	5.72 5.99	6.86	0.53	0 75 0.79	0.90	0.60	0.40	0.35	0.25	0.22	0.20	5.38	7.09 7 40	8.31	16.570	21.837 22.792	25.595
25	1.50	2.06 2.18	2.44	3.43	4.71 4.99	5.58	.45	.62 .66	.73	.60	.40	.35	.25	.22	.20	4.73	5.95 6.27	6.86	14.568	18.326 19.311	21.129
50	1.25	1.63 1.75	1.88	2.86	3.73 4.00	4.30	.38	.49 .53	.56	.60	.40	.35	.25	.22	.20	4.09	4.84 5.15	5.41	12.597	14.907 15.862	16.663
75	1.00	1.19 1.31	1.31	2.29	2.72 3.00	3.00	.30	.36 .39	.39	.60	.40	.35	.25	.22	.20	3.43	3.72 4.01	3.94	10 564	11.458 12.351	12.135
100	.75	.75	.75	1.72	1.72	1.72	.23	.23	.23	.60	.40	.35	.25	.22	.20	2.80	2.57	2.50	8.624	7.916	7.700
Lbs. of Ex.-steam Used per I.H.P. per Hour.	Same as Above Except that Weights of Exhaust Steam are Used Instead of Per Cents.																				
0	1.75	2.50	3.00	4.00	5.72	6.86	0.53	0.75	0.90	0.60	0.40	0.35	0.25	0.22	0.20	5.38	7.09	8.31	16,570	21,837	25,595
5 625	1.40	2 00	2.44	3.20	4.58	5.58	.42	.60	.73	.60	.40	.35	.25	.22	.20	4.47	5.80	6.86	13,768	17,864	21,129
11.25	1.15	1.55	1.88	2.63	3.55	4.30	.35	.47	.56	.60	.40	.35	.25	.22	.20	3.83	4.64	5.41	11,796	14,391	16,663
16.875	.93	1.15	1.31	2.13	2.63	3.00	.28	.35	.39	.60	.40	.35	.25	.22	.20	3.26	3.60	3.94	10,041	11,088	12,315
22.5	.75	.75	.75	1.72	1.72	1.72	.23	.23	.23	60	.40	.35	.25	.22	.20	2.80	2.57	2.50	8,624	7,916	7,700

These tables show the running-expense independent of the first cost and maintenance of plant.
The double figures for condensing-engine are arrived at as follows:

No exhaust-steam used...	Full condensing. [illegible] "	50 per cent exhaust-steam used........	[illegible] condensing. [illegible] "
25 per cent " "	[illegible] " [illegible] "	75 " " " " " .... 100 " " " " "	[illegible] " Full high pressure.

COST OF PLANT PER I. H. P. TO CHARGE TO POWER, AND TOTAL YEARLY EXPENSE PER I. H. P.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Engine.	Per Cent of Exhaust-steam used.	Engine and Piping complete.	Engine-house.	Engine Foundations.	Total Cost of Engine-plant.	Depreciation at 4% on Total Cost.	Repairs at 2% on Total Cost.	Interest at 5% on Total Cost.	Taxation at 1.5% on ¾ Cost.	Insurance at 0.5% on Engine and Engine-house.	Totals of Cols. 7, 8, 9, 10, and 11.	Boilers Complete, including Feed-pumps, etc.	Boiler-house.	Chimney and Flues.	Total Cost of Boiler-plant.	Depreciation at 5% on Total Cost.	Repairs at 2% on Total Cost.	Interest at 5% on Total Cost.	Taxation at 1½% on ¾ Cost.	Insurance at 0.5% on Total Cost.	Totals of Cols. 17, 18, 19, 20, and 21.	Total Ordinary Running Expense, Cols. 20, 21, and 22.	Total Yearly Expense per I. H. P. Col. 12 + 22 + 23.
		$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$
Compound.	0	25.00	8.00	7.00	40.00	1.60	0.80	2.00	0.45	0.165	5.015	9.33	2.92	6.11	18.36	0.918	0.367	0.918	0.207	0.092	2.502	16.570	24.087
	25											8.00	2.50	5.66	16.16	.808	.323	.808	.182	.081	2.202	14.568	21.785
	50	Constant.	"	"	"	"	"	"	"	"	"	6.67	2.08	5.15	13.90	.695	.278	.695	.156	.070	1.894	12.597	19.506
	75											5.33	1.67	4.60	11.60	.580	.232	.580	.131	.058	1.581	10.564	17.160
	100											4.00	1.25	4.00	9.25	.463	.185	.463	.104	.046	1.261	8.624	14.900
Condensing.	0	20.00	7.50	5.50	33.00	1.32	.66	1.65	.371	.138	4.139	13.33	4.17	7.30	24.80	1.240	496	1.240	.279	.124	3.379	21.837	29.355
												13.97	4.37	7.46	25.80	1.290	.516	1.290	.290	.129	3.515	22.792	30.446
	50	20.00	7.50	5.50	33.00	1.32	.66	1.65	.371	.138	4.139	10.99	3.43	6.70	21.12	1.056	.422	1.056	.233	.106	2.878	18.326	25.343
												11.62	3.63	6.84	22.09	1.106	.442	1.106	.248	.111	3.013	19.311	26.463
	25	19.00	7.50	5.00	31.00	1.26	.63	1.575	.354	.133	3.952	8.69	2.72	5.92	17.33	.867	.347	.867	.195	.086	2.362	14.907	21.221
												9.33	2.92	6.11	18.36	.918	.367	.918	.207	.092	2.502	15.862	22.316
	75	19.00	7.50	5.00	31.00	1.26	.63	1.575	.354	.133	3.952	6.35	1.98	4.92	13.25	.663	.265	.663	.149	.066	1.806	11.458	17.216
												6.99	2.18	5.15	14.32	.716	.286	.716	.161	.072	1.951	12.351	18.254
	100	*																					
Non-condensing.	0	17.50	7.50	4.50	29.50	1.18	0.59	1.475	0.332	0.125	3.702	16.00	5.00	8.00	29.00	1.450	.580	1.450	.326	.145	3.951	25.595	33.248
	50											13.01	4.07	7.20	24.28	1.214	.486	1.214	.273	.121	3.308	21.129	28.139
	25											10.03	3.13	6.30	19.46	.973	.392	.973	.219	.097	2.654	16.663	23.019
	75	Constant.	"	"	"	"	"	"	"	"	"	6.99	2.18	5.30	14.47	.724	.289	.724	.163	.072	1.972	12.185	17.859
	100											4.00	1.25	4.00	9.25	.463	.185	.463	.104	.046	1.261	7.700	12.663

* The condensing-engine at 100 per cent of exhaust-steam used becomes a non-condensing engine.

YEARLY SAVING BY COMPOUNDING SIMPLE ENGINES.

Horse-power.	Saving in Fuel per Year of 309 Days of 10 Hours each.	Horse-power.	Saving in Fuel per Year of 309 Days of 10 Hours each.
200........	$620.75	800.....	2,483.03
250........	775.94	900......	2,793.41
300........	931.13	1,000......	3,103.79
350........	1,086.32	1,100......	3,414.17
400........	1,241 51	1,200......	3,724.55
450........	1,396.70	1,300......	4,034.93
500........	1.551.89	1,400......	4,345.31
600........	1,862.27	1,500.	4,655.69
700........	2,172.65		

the train as short, and the power expended in their operation as small, as is practicable. Small cost of construction must usually be made a consideration secondary to that of maintenance and operation.

Occasionally, special considerations may prove imperative, and may seriously modify what would otherwise be the engineer's plan. Thus, the introduction of a steam-plant into a populous neighborhood in a city, or into a large and valuable or a crowded building, may compel him to use some form of "safety steam-generator," where he would otherwise adopt a cylindrical tubular boiler. The advisability of employing several small detached engines, rather than one large one, may lead him to choose the high-speed type rather than a moderate-speed engine of the older kind; objections to the rumble of gearing may induce him to use belting for transmission of large amounts of power; or an exceptionally low speed of machinery may compel him to employ gearing where he would prefer to use belts or rope-transmission. In every case the broader the view which the designing engineer is able to take of the problem in hand, the more certain is he to effect a satisfactory and the best solution.

The "Steam-plant," so called, thus consists of the whole arrangement of engines, boilers, and all accessories, and its design frequently involves many important questions arising from the conditions of location, and of costs at the chosen locality.

In the figure, these have been studied with a view to securing the best and most convenient location of parts, actually and

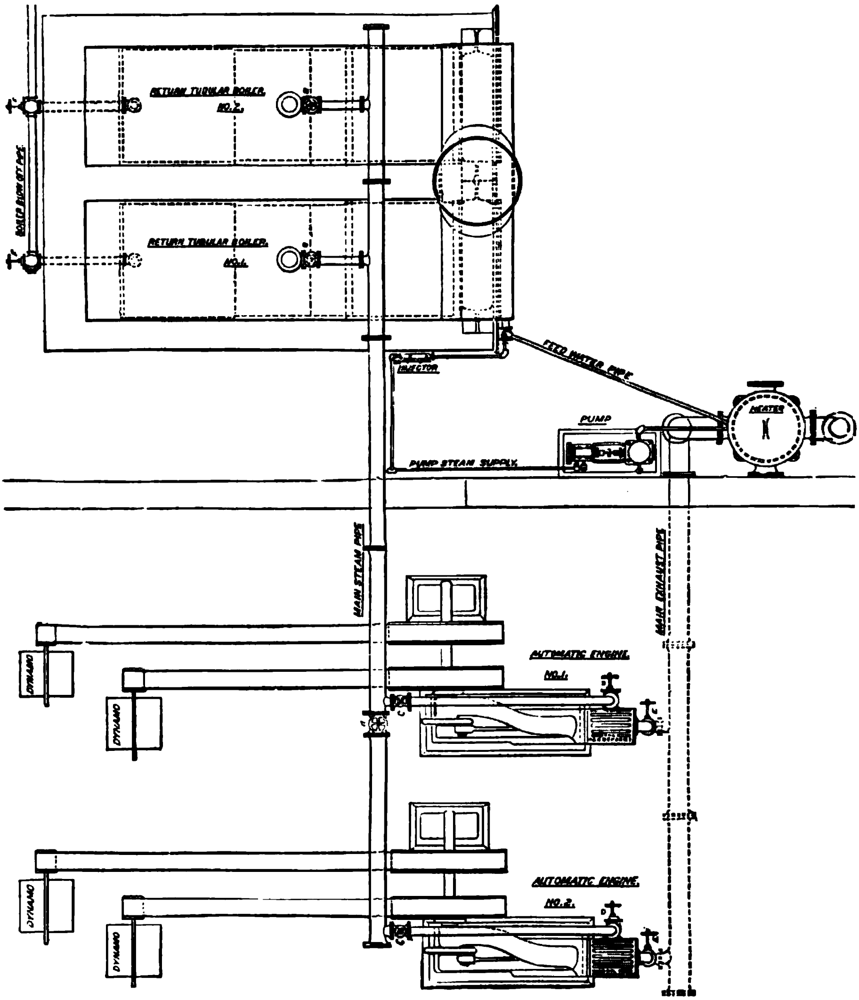

FIG. 216.—THE PLAN OF THE "PLANT."

relatively; to give minimum costs of transportation of fuel and ash, short steam- and exhaust-pipes, limited floor-space, and the closest possible connection of the engines with their work—in

this case the driving of dynamo-electric machinery—both for economical reasons, and to enable the oversight of the attendants to be thoroughly efficient.

244. **Electrical Energy** is distributed, as a rule, from a single station to all parts of a city or town, except in the case of the very largest cities, where it is found necessary to divide into districts of carefully studied extent, each having its own station. The difficulty of designing such a system as shall in each case prove to be the correct solution of the problem of maximum commercial efficiency, may be imagined when the demand for power and currents, hour by hour, is graphically illustrated, as in the figure. The demand is constantly varying, and through an enormous range. Throughout the whole range of power and time, maximum commercial efficiency demands a constantly varying apportionment of power in the engine and variable number and efficiency of engines. The engineer must therefore seek a mean which shall give a final maximum resultant efficiency.

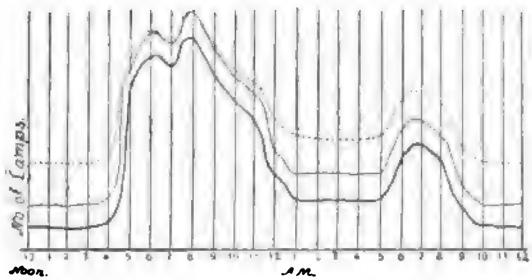

FIG. 219.—DISTRIBUTION OF ENERGY.

The motive-power must in all such cases be apportioned, and the sizes and kinds of engine adopted, only after a very careful and minute study of the anticipated or actual demand and all its variations, so far as it is possible to anticipate them for the immediate future, if not for the life of the "plant." It may be found that a large engine of the most perfect and elaborate construction, to secure great economy of fuel, should be supplemented by a number of other small, inexpensive, and

wasteful machines, and even that it will pay well to make these of various sizes and degrees of excellence, precisely as it may be found desirable to go further into detail and assign to each, and to each principal part of each, a specified quality of lubricant.

Actual costs of every detail, of attendance, fuel, oil, repairs, interest, rents, and taxes, must be either known or carefully estimated, hour by hour and year by year, for every part.

But it is important that the designer keep in mind the fact that the first and most imperative condition of ultimate success and economy is absolute reliability at all times and under all reasonably conceivable conditions. This often dictates the free purchase of spare machinery and of spare parts. Careful provision for growth is usually important, and such provision is a condition favorable to the last-mentioned provision.

Whether one large and comparatively economical engine, or several small and more wasteful machines, or an intermediate system, should be adopted, will be determined by the anticipated magnitude and variations of the load. The larger the demand and the more uniform, the more nearly are the conditions suitable for the first; the less the total power and the more variable, the better is the second suited to the case. In making a determination of this question, the designing engineer must consider the risks involved, on the one hand, of disaster from the remote probability of accident disabling the large engine; on the other hand, of a less serious result following the greater probability of a breakdown of one among several smaller engines. To a certain extent, also, this problem presents itself in the assignment of the number and the individual power of the dynamos employed. As respects the engines, a loss of power, probably, of ten per cent on directly-connected engines, and of one fourth or one fifth with those indirectly connected, must be counted on; but, within ordinary limits of size, the efficiency of the dynamo should be nearly constant.

The elements of the problem are illustrated in the sketch already given, in which Mr. Bryan exhibits the work at the

lamps, that work plus friction, and, in the upper line, the equivalent of the heat-energy of the steam used for a period of twenty-four hours.

The choice of boiler is also determined by the principles which have now been so fully enunciated. It is impossible to present a permanently valuable specific example, since values are constantly and greatly changing. Each case must be worked up by itself, and with reference to the costs current at the time and location of the construction.

Mr. Field gives the following facts relative to best current practice (1890–91) in the installation of electric systems of power-transmission:

The cost of an electric-car equipment, including two motors, truck and car-body complete, is from $3200 to $3500.

There should be installed, in generating-capacity for power-plant, twenty to twenty-five horse-power per car operated, which will give reserve power.

The cost of generating power is from three to five cents per car-mile.

A car uses under average conditions one H. P. per car-mile per hour. That is, a car operating at a speed of five miles, five H. P.; at eight miles, eight H. P.

Cars are generally equipped with two fifteen-H.-P. motors. The tendency at present time is to slower-speed motors, thereby reducing amount of gearing.

The attainable speed with electric motors is limited only by condition of road-bed and local requirements; 130 miles an hour has been attained experimentally.

Electric traction means rapid transit and increase of traffic of from 40 to 200 per cent, and moderate reduction in operating expenses per car-mile.

One mile of single-track construction will cost complete, with 65-pound girder-rail, ties two and one half feet on centres, bonding of rails, paving, etc., $9000 to $10,000.

The cost of the electric part of power-plant, including generators, switch-board, etc., installed, is $35 to $45 per H. P.

Line-construction per mile, complete, including track-bonding, plain pole-work, cross-suspension or bracket with feed-wire	$2,000	to	$2,500
Sawed and painted poles	2,500	to	3,000
Iron poles, concrete setting, cross suspension, double-track, feed- and guard-wires	6,500	to	7,500
Same with centre poles	4,500	to	5,500

An electric car averages 100 to 125 miles a day.

Engines operate at a piston-speed of 600 to 800 feet per minute, high-speed—so called—having in general the lower piston-speed and shorter stroke and higher rotative speed or number of revolutions, and the Corliss the reverse. Corliss engines generally operate at 75 to 100 revolutions, and larger high-speed 150 to 225 revolutions.

Belt-generators direct to engines with Corliss and high-speed, thus making each unit independent of all others, one generator to each engine preferable.

Avoid use of counter-shafting by above methods.

Make belt-centres as long as practical; 20 to 30 feet for high-speed, and 40 to 50 feet with Corliss.

Engines on railway work should be built very heavy, and have ample fly-wheel capacity to relieve the working parts of the excessive strains due to changes in load.

Compound engines should not be run non-condensing on railway work; the loads are too variable.

Every effort should be made in locating powerp-lant to obtain facilities for condensing, and operate plant with compound condensing engines.

Ten to thirteen square feet of heating-surface, evaporating 30 lbs. of water per hour is the usual unit of H. P. for boilers.

Compound condensing engines require only half of the boiler capacity of single-cylinder ones, and give a corresponding economy in coal-consumption.

On the designing and arrangement of power-stations depend the economy of operation, both for present requirements and future developments.

Best practice is tending to direct coupling of engine and generator.

CAPACITY OF ENGINE REQUISITE FOR DIFFERENT GENERATORS.

Generator.	Engine.						
		High Speed.			Corliss.		
Watts.	H. P.	Size.	Speed.	Weight two Fly-wheels.	Size.	Speed.	Weight two Fly-wheels.
50,000	75	12×12	280	7,000 lbs.			
80,000	125	15×16	225	9,000 "			
150,000	225	18½×18	200	15,000 "	20×36	90	25,000 lbs.
2–150,000	450				24×48	80	50,000 "

Steam-pressure 100 lbs.

COST OF ELECTRIC EQUIPMENTS FOR STREET-RAILROADS.

No of Cars.	Steam-plant, H. P.	Capacity of Generators K. W.	Steam-plant.	Station Electrical Equipment.	Car-equipments, Boilers, Trucks, and Motors.	Line-construction half mile of Double-track per car.	Total Equipment (omitting Track).
6	120	80	$7,000	$6,400	$19,500	$15,000	$47,900
10	225	150	11,000	10,500	32,500	25,000	79,000
15	375	240	17,500	15,000	48,750	37,500	118,750
20	450	300	22,000	17,500	65,000	60,000	164,500
30	675	450	28,000	22,000	97,500	90,000	237,500
50	1125	750	50,000	33,000	162,500	187,500	433,000
100	2025	1350	90,000	60,000	325,000	375,000	850,000

The following estimates will illustrate a fair distribution of costs of electric-power distribution in various quantities: *

ELEMENTS OF THE COST OF ONE H. P. HOUR (ELECTRIC) IN CENTS.

Capacity in Elect. H.P.	100	300	500	800	1000	1500	2000	3000	4000	5000	6000
Engineer	0.4	0.13	0.08	0.05	0.04	0.04	0.04	0.04	0.04	0.04	0.04
Fireman	0.3	0.10	0.06	0.037	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Dynamo-man	0.4	0.13	0.08	0.05	0.04	0.04	0.04	0.04	0.04	0.04	0.04
Helper	0.25	0.08	0.05	0.031	0.025	0.025	0.025	0.025	0.025	0.025	0.025
Superintendence	0.30	0.10	0.06	0.037	0.03	0.02	0.015	0.001	0.001	0.001	0.001
Coal	0.475	0.475	0.475	0.475	0.475	0.475	0.475	0.475	0.475	0.475	0.475
Oil, waste, and water	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15
Interest and depreciation steam-plant	0.057	0.051	0.044	0.033	0.028	0.022	0.022	0.022	0.022	0.022	0.022
Interest and depreciation, electric plant	0.057	0.051	0.044	0.033	0.028	0.022	0.022	0.022	0.022	0.022	0.022
Interest and depreciation, building	0.028	0.026	0.022	0.168	0.014	0.011	0.011	0.011	0.011	0.011	0.011

* Trans. Am. Inst. Elect. Engrs., 1890.

The cost of distribution of electrical energy in moderate amount may be exemplified by the following estimates by Mr. Scheffler.*

The generating-station contains a 360 horse-power engine, and sufficient boilers for operating the same. The cost of the station can be summed up about as follows:

1300 horse-power engine	$10,000
Battery of 360 horse-power boilers	7,000
Station-building	5,000
Steam-connections, pumps, feed-water heaters, etc	2,000
Generating dynamo	12,000
Electrical station appliances, etc	1,000
Making a total of	$37,000

The foregoing station outfit will not permit of having an auxiliary plant of engine, dynamos, and boilers, which ought really to be included in the outfit, so that in case the engine or dynamos or boilers should at any time become inoperative there would be another set of appliances to operate the railroad with. An auxiliary set of appliances would almost double the total cost of the station.

For operating expenses of the station we have the following:

5 per cent depreciation on engine and boilers	$835
4 " " " " dynamo	480
6 " " interest on station-plant	2,220
Two firemen, night and day	1,440
Two engineers, night and day	2,000
Two laborers, night and day	1,080
Maintenance, such as oil, waste, etc	300
Making a total of	$8,355

* Trans. A. S. M. E., 1890.

per year, not including the cost of coal, which may be reckoned at about 2800 tons per annum, at 3 pounds per horse-power per hour; while, if applied to the operation of a street-railway, the out-of-door expenses would amount to some $25,000 additional, up to the point at which the current was received by the motors. The additional cost of cars and their operation would vary greatly with local conditions.

In the electric distribution of power it is usually found that the average load on the electric motor is less than one half its rated power, and that the maximum load seldom exceeds this average by one third, or equals two thirds the capacity of the machine; so that a motor of a given rated power being employed at $5 per month per horse-power, as rated, usually actually pays the station about $10 for power actually delivered. Where, as is sometimes the fact, the charge is for a rate indicated by the maximum reading on the "ammeter," the real payment thus becomes about $7, or less than the cost of power from a gas-engine consuming 20 cubic feet per hour per horse-power, at $1.50 per thousand ($7.75).

The following were the rates charged for electric power in several large cities in the year 1890: *

NEW YORK.

Intermittent Work.	Per Month.	Freight Elevators. Per Month.
1/8 H. P.	$3 00	......
1/4 "	5 00	......
1/2 "	8 00	......
1 "	12 50	$10 00
2 "	22 50	15 00
3 "	30 00	18 33
4 "	36 66	21 66
5 "	43 75	25 00
6 "	50 00	28 33

* Transmission of Electric Power; C. and C. Co., 1891.

7 H. P.	56 88	31 66
8 "	63 33	35 00
9 "	69 37	38 33
10 "	75 00	41 66
15 "	100 00	60 00
20 "	125 00	......
25 "	145 83	......

Based on maximum readings.

Continuous work, 50 per cent extra.

Meter Rate, 10 cents per H. P. per hour, with discounts as follow:

	Horse-power. Hours per Month.	
On bills for	100 to 200............	20 per ct.
" "	200 to 400............	25 "
" "	400 to 600............	30 "
" "	600 to 800............	35 "
" "	800 to 1,000............	40 "
" "	1,000 to 1,500............	45 "
" "	1,500 or over............	50 "

BOSTON, MASS.

10 cents per H. P. per hour, with discounts from monthly bills as follow:

Bills of from	$5 00 to $8 00..........	20 per ct.
" "	8 00 to 13 00..........	25 "
" "	13 00 to 20 00..........	30 "
" "	20 00 to 30 00..........	40 "

The rate for 10 H. P. would be a similar scale with the following discounts:

Bills of from	$30 00 to $44 00.......	20 per ct.
" "	44 00 to 60 00........	25 "
" "	60 00 to 95 00........	30 "

ROCHESTER, N. Y

(WATER-POWER.)

⅛ H. P............	$18 00 to $36 00	per year.
½ "	5 00	per month.
1 "	7 00	"
2 "	10 00	"
5 "	25 00	"
10 "	36 00	"
15 "	50 00	"

CHICAGO, ILL.

Applicable to classes of work in which the rate of power-consumption is uniform, or nearly so, such as operating ventilating-fans.

	Per Month.		Per Month.
¼ H. P........	$5 00	7 H. P.........	$65 45
½ "	8 50	8 "	73 20
1 "	14 00	9 "	81 45
2 "	25 00	10 "	90 00
5 "	50 75	15 "	135 00
6 "	57 90		

Applicable to classes of work in which the rate of power-consumption is variable, such as operating machinery for printing, sewing, embossing, wood and metal working, pumping for hydraulic elevators, coffee-grinding, baking, etc.

	Per Month.		Per Month.
⅛ H. P........	$3 00	6 H. P.........	$42 60
¼ "	4 50	7 "	47 25
½ "	6 50	8 "	52 50
1 "	10 00	9 "	57 15
2 "	18 50	10 "	62 50
5 "	37 25	15 "	93 75

Based on maximum readings.

Elevator Work, applicable only to elevators driven by belts direct from the motor.

	Per Month.		Per Month.
4 H. P.......	$26 00	8 H. P.........	$42 00
5 "	30 00	9 "	46 00
6 "	34 00	10 "	50 00
7 "	38 00	15 "	70 00

DES MOINES, IOWA.

(WATER-POWER.)

	Constant. Per Month.	Intermittent. Per Month.
¼ H. P......................	$4 00	$3 00
½ "	5 00	3 75
1 "	8 00	6 00
2 "	14 00	10 00
3 "	20 00	14 00
4 "	25 00	17 00
5 "	30 00	20 00
7½ "	40 00	28 00
10 "	50 00	33 00
15 "	70 00	46 00
20 "	90 00	60 00
25 "	110 00	72 00

Elevator Work.

	Per Month.
3 H. P..............................	$12 50
5 "	18 00
7½ "	25 00
10 "	30 00
15 "	40 00

The actual costs, in the case of a large modern establishment for electric, lighting, designed by Mr. Field, were the following:

COST OF PLANT.

Present capacity for generating	12,000—16 c. p.	
" " underground	20,000 " "	
Ultimate " for generating	36,000 " "	
Station building, including fittings, foundations, stacks, furniture, office fittings, etc., for ultimate		$100,000
Real estate for building 75×100		36,000
Steam-plant, including engines, boilers, pumps, heaters, piping, belts, water-tanks, coal and ash handling, etc. (12,000 H. P.)		50,000
Electrical plant, including dynamos, switch-board, cables, etc		40,000
Underground materials, as tubing for mains and feeders, junction-boxes, joints, etc		115,000
Installation of same, including trench-work, freight, cartage, laying, etc		35,000
General sundries, as lamps, meters, tools, instruments, engineering, and superintendence, etc		50,000
		426,000

Cost of completing Plant.

Steam-plant	80,000	
Electrical plant	65,000	
Underground	300,000	
General sundries	75,000	
Total cost of complete plant		520,000
		946,000

Summary of Cost.

* Steam-plant per H. P	$37 00
Electrical plant per 16 c. p	3 00
" " " E. H. P	30 00
Underground " 16 c. p	7 50
Total cost per H. P. output	270 00
" " " 16 c. p. "	27 00

A recent estimate by Mr. Kapp gives the following as the maximum distances of energy-transmission at the date (1890) for the specified systems:

Power and Source.	Distance (max.) at 90 per cent Efficiency of Transmission.		
	Highway.	Tramway.	Miles Rail.
Steam-engine and coal	115	270	1300
Horse; corn	52	170	440
Electric motor and storage-battery	4	10	26

* Foundations and boiler-settings included in building.

The cost of steam-power is here reckoned at \$50 (£10) per H. P. per annum, 3000 hours' work. These figures are, however, continually changing, and may probably be expected to steadily become more favorable to the use of electricity.

Kapp gives the following expression for the current of maximum economy in power-transmission:*

Let D = distance in miles;
a = section of conductor in square inches;
E = terminal volts at generator;
e = terminal volts at motor;
HP_g = B. H. P. required to drive generator;
HP_m = B. H. P. obtained from motor;
c = current in amperes;
(Efficiency of generator, 90 per cent; efficiency of motor, 90 per cent;)
g = cost per electrical H. P. output of generator;
m = cost per B. H. P. output of motor, including regulating gear;
$G = .9gHPg$ = cost of generator;
$M = mHP_m$ = cost of motor and regulating gear;
$t = 18.2Da$ = weight in tons of copper in line;
K = cost per ton of copper, including labor in erection;
s = cost of supports of line per mile run;
p = cost of one annual B. H. P. absorbed by generator;
(Costs are taken in British money;)
q = percentage for interest and depreciation on the whole plant.

$$\text{Capital outlay} = gEc/746 + mHP_m + Ds + \frac{1.6KD^2c^2}{Ec - 830HP_m} = A.$$

$$\text{Annual cost per B. H. P. delivered} = qA/HP_m + pHP_g/HP_m.$$

* Cantor Lectures, delivered before the London Society of Arts, February 1891.

Put $B = Ep/670 + qEg/746$.

$$\gamma = 830HP_m/E,$$

the current which would be required if the line had no resistance, and

$$\beta = \gamma^2 \frac{EB}{1.6qKD^2 + EB};$$

then the most economical current at the given voltage E is

$$c = \gamma\left(1 + \sqrt{1 - \frac{\beta}{\gamma^2}}\right),$$

$$c = \gamma\left(1 + \sqrt{\frac{1.6\,qKD^2}{1.6\,qKD^2 + BE}}\right).$$

For very long distances the term under the square-root approaches unity, and the most economical current the value 2γ; from which it follows that under no circumstances will it be economical to lose more than half the total power in the line.

The conditions on which the formula is based are:

Annual value of brake horse-power at generating station; voltage at generator terminals; brake horse-power required at motor end; distance of transmission; cost per horse-power of machines and regulating appliances at the given output, and voltage; cost of conductor per ton of copper erected, interest, and depreciation of whole plant.

The data required: Working current; brake horse-power at generating station; mechanical efficiency; voltage at motor; total capital outlay per brake horse-power delivered; and cost of annual brake horse-power.

The efficiency of each machine is assumed to be 90 per cent. The formula gives the current, and the other data can be found therefrom. The cost of supports for the line per mile, whether overhead or underground, may be taken as constant, and it therefore does not enter into the formula. Interest and depreciation are taken the same for all parts of the plant, to avoid complication. We thus find a voltage for

which the annual cost of the brake horse-power, delivered at the motor end of the line, is a minimum. The greater the cost of power at the generating station, the higher is the economical voltage, this voltage also increasing with distance.

Each case must be worked out with due regard to local conditions.

It is necessary to take into account the cost of the power, both at the generating and at the receiving stations; the interest and depreciation of the line, and the interest and depreciation of the machinery at either end; and in estimating these items it must be known at what voltage the plant is to work, and what total power is required. The prime cost per horse-power depends on total power and voltage.

About 0.70 of the average total power supplied at the engines is lost in the average case on the shafting. Motors have a working efficiency ranging from 60 to 85 per cent, and averaging about 0.75.

The Oerlikon Works, Switzerland, supply the following figures, which give the whole outlay for electrical parts of power-transmissions erected by that establishment:

COST OF ELECTRICAL TRANSMISSION.

Distance in Miles.	H.P. Delivered.	Revolutions per Minute.	Cost.			Total Cost.*	Cost per H.P.
			Gen.	Motor.	Line.		
1.870	85	450	640	560	440	1,880	22.2
.280	195	500	760	680	132	1,800	9.7
.280	51	600	320	280	60	720	14.1
.375	90	550	520	480	80	1,240	13.8
.560	71	600	440	400	60	1,040	14.6
.280	40	700	260	240	20	640	16.
.375	75	600	480	440	68	1,120	15.
.500	87	500	520	480	100	1,260	14.5
1.560	150	600	760	720	330	2,050	13.7
.220	93	450	440	420	232	1,270	13.7
6.250	11	900	132	110	480	960	87.
2.200	51	600	360	320	300	1,140	22.4
.187	60	900	240	220	18	600	10.
5.000	41	750	240	200	344	1,020	24.8
3.750	220	600	1,040	960	640	2,960	13.5
.002	15	600	112	104	8	252	16.8
.250	19	700	160	160	20	390	20.5

* Including all accessories, and reckoned in British money.

The following were fair average figures in 1890, but are subject to continual variation with the state of the market:

COST OF STEAM AND ELECTRICAL PLANTS—COST OF ENGINES FOR SIZES OVER 100 H. P.

High-speed, single	$11	to	$13	per	H. P.
" " compound	14	"	16	"	"
Corliss, single	16	"	18	"	"
" compound	22	"	25	"	"
" triple	27	"	30	"	"

COST OF STEAM-PLANTS,

Including Engines, Boilers, Piping, Pumps, Heaters, Foundations, Settings, etc.

High-speed	$40	to	$50	per	H. P.
Corliss, direct connection	60	"	70	"	"
" counter-shaft	80	"	85	"	"

COST OF ELECTRICAL PLANT.

Dynamos, switch-board, cables, foundations, erecting, etc.	$35	to	$40	per	H. P.
Pole-line, including mains, feeders, poles, setting, etc.	4	"	6	"	16 C. P.
Underground ditto	7	"	9	"	" "
Inside wiring, including lamp, socket, plain pendant and rubber-covered wire, moulded work	4	"	5	"	" "
Same, for concealed work	5	"	6	"	" "

Mr. Field gives the following figures (1891) for a case of the purchase, equipment, and operation of a street railway system with electricity, a city with a population of, say 100,000—with a dilapidated street railway system, earning a gross income of $125,000, to purchase same for $500,000—property rights, franchises, etc., and equip it with 40 miles of single track and 65 electric cars: *

* Trans. Nat. El. Lt. Assoc., Montreal Meeting, 1891.

COST OF EQUIPMENT.

Steam-plant (1500 H. P. steam-plant):		
Five engines, 250 H. P. each, compound condensing, size 16 inches × 32 inches × 42 inches, with wheels weighing 30,000 pounds	$32,500	
Eight R. T. boilers, 72 inches × 16 feet	9,600	
Jet condensers	3,000	
Two boiler feed-pumps	900	
Steam and exhaust piping	12,000	
Five engine-foundations	3,500	
Eight boiler-settings	3,200	
Five 30-inch belts	2,000	
Erecting and starting	3,500	
Freight and miscellaneous	2,500	
		$72,700
Electrical plant:		
Five generators, 200 kilowatts	$37,500	
Switchboard installation, foundations, etc.	4,000	
		41,500
Building:		
Power-station, including stack, travelling crane, etc.	$25,000	
Car-house and repair-shop, including tools, etc.	15,000	
		40,000
Track-construction:		
40 miles girder-rail construction, ties 2½ feet centres, 63-lb. rail, etc., $1.15 per foot	$242,880	
Relaying, including paving, etc., at 60 cents per foot	126,720	
Trucking, hauling, etc.	24,000	
Ties, including 10 per cent of joint ties, 130,000 at 40 cents	52,000	
Ties, including 10 per cent of joint ties, 15,000 at 70 cents	10,500	
		456,100

Line-construction:			
Ten miles iron poles, etc.		$75,000	
Ten miles wooden poles, etc.		40,000	
			115,000
Car-equipment:			
65 electrical equipments at $2,000		$130,000	
65 car-bodies, 18-foot body, with open ends		65,000	
65 trucks at $250		16,250	
			211,250
Summary:			
Steam-plant	$72,700		
Electrical plant	41,500		
Building	40,000		
Track	456,100		
Line-construction	115,000		
Car-equipment	211,250		
		$936,550	
Superintendent's and Engineer's work	$50,000		
General and miscellaneous	50,000		
		100,000	
		$1,035,550	
Original purchase		500,000	
Total cost re-equipped			$1,535,550

Gross income, say, $350,000.

The transmission of power over very great distances, as from a waterfall or prime motor to a town or an establishment several miles away, is best effected by the electric current at high tension, as to London from Greenwich by the Ferranti system, or from the Neckar Falls at Dauffen to Frankfort, Germany. The latter is about 110 miles (180 kilometers) and a "pressure" of 25,000 volts is adopted by its designers, the Oerlikon Works, and 300 electrical horse-power is transmitted.

In the organization of such a system of distribution the division of duties is somewhat as follows: *

* Trans. Nat. El. Lt. Assoc.; 1890.

The superintendent is in charge of the station, and of all work. If the station is large enough, he may have a man who can attend to the making out of reports. His assistant takes his place at times, and can often decide whether work shall be done, or whether it shall wait for the superintendent's return.

Under the superintendents are the chief engineer, chief dynamo-man, chief lineman, and chief incandescent wireman.

Under the chief dynamo-man are various dynamo-runners, although the chief engineer may be able to take charge of the dynamo-room.

Under the chief engineer are all engineers, foremen, coal-handlers, etc.

In the absence of the chief engineer, the engineer on watch takes his place, and the same with regard to the dynamo-man.

The chief lineman should have under his care pole-lines, outside construction of all kinds, including all arc-lamps and high-tension wires; also the care of converters, if any, to the first cut-out on the secondary side. He should also have charge of the carbon-setters and arc patrolmen.

The chief wireman should have charge of inside wires, all lamp-renewals, patrolmen, and material and work.

The storekeeper has duties separate from all these.

The reports necessary are the engineer's log, for which there should be two books provided,—one for the day run, and one for the night run,—to be filled up by the engineer on watch and turned into the superintendent's office each day for examination.

The dynamo-room log should include a reading of the load on each machine, taken at twenty-minute intervals throughout the whole run. This report, with the amount of coal burnt, gives a check on the fireman and the quality of coal.

The inspectors and patrolmen should fill out a form, giving the number of arc-lamps on the circuits; if any are out or burning badly, between what hours, and the probable cause.

The storekeeper should make a report daily of all material received and issued, which can be used as a check upon bills for material.

245. **The Costs of Distribution of Power** include, ordinarily, those of overcoming the friction of a large amount of shafting and belting. The magnitude of this cost has been carefully studied by Mr. Henthorn.* The power demanded in cotton and woollen mills is rarely much, if any, less than 20 per cent of that furnished by the engine, while it sometimes amounts to 30 per cent and over.

The transmission of steam-power, and the application of energy at a distance from the primary source, with economy, safety, and certainty, may often prove a problem of such importance as to justify a careful study and comparison of all available methods. These methods include—

(1) Transmission of energy by carriage of steam to motors distant from the boilers.

(2) Transmission of power from engines set beside the boilers.

In the former case the total power may be supplied, often, either from a single engine or by supplying the steam to a number of smaller engines distributed as may be found best in facilitating work; and the problem includes that of determining what arrangement aud distribution of these engines is, on the whole, most desirable. In the latter case the problem includes the comparison of various methods of transmission of energy from the engines to the work.

The transmission of steam is rarely practised over distances exceeding a few hundred feet at most; although there are no insuperable difficulties, ordinarily, to carrying it thousands of feet, the steam-pipes being well clothed, kept dry, and thoroughly and automatically drained at all low points by traps, separators, or other system. With intermittent service, however, this method is usually found both wasteful and troublesome, especially in meeting the variations of length due changing temperatures, and the "water-hammer" liable to occur when steam is introduced into cold pipes. External drainage of the trenches in which the pipes may be laid is quite as important as the internal drainage of the pipes themselves.

* Trans. Am. Soc. M. E.; vol. VI., No. CLXXVII.

Distances approximating a half-mile are thus readily attained. The higher the steam-pressure maintained in the pipes in any given case, the less the loss of pressure by friction, and, usually, the less important any stated drop of pressure.

The engines being placed at the boilers, the transmission of their power to their work may take place by either of several ways:

(1) By shafting and pulleys.
(2) By wire ropes.
(3) By water-pressure in pipes.
(4) By compressed air.
(5) By electric currents.

For short distances, as within workshops as commonly constructed, lines of *shafting* are safest, cheapest, and in all ways best. For driving large isolated and widely distributed machines, it is often better to adopt a small engine for each or at each group, supplied with steam from the main boilers, with water or air under pressure from pumps driven by the main engine, or with motors driven by currents supplied from electro-dynamic generators similarly driven. The limit for shafting, as an average, may be taken as about 1000 feet; but the coëfficient of friction of shafting, under its light pressures, is very great; the friction is intensified and power wasted by the often inevitable "getting out of line," and the weight is considerable, so that the total waste is apt to be serious. In such cases the engine distributing its power should be as near the centre of power utilization as possible.

All pulleys should be carefully balanced and should "run true." All others should be promptly condemned. Tubular shafting has the advantage of stiffness, the disadvantage of large friction. Belting is used under all ordinary conditions; but it is unfitted either for the transmission of exact velocity-ratios or for slow speeds of rotation and large power.

Where the span is considerable and no shafting is desired, ropes of hemp, cotton, or wire are often employed in place of belts, and 50 H.P. per rope, if hemp or cotton, of 7 inches circumference, at 1500 feet per minute, is considered good

practice. With such ropes arranged "in multiple," great care must be exercised to see that their pulleys are precisely alike in size. Chains may be used instead of belting for very slow and heavy work.

Wire-rope transmission is employed very extensively for long distances, the carrying pulleys being set at distances of 200 to 500 feet apart, and made of a diameter usually not less than 100 times that of the rope, and preferably 150. It is found that distances of 10 miles or more may be thus attained with a loss not exceeding 25 per cent.

Water-pressure of great intensity, with flow at a moderate rate, often proves a satisfactory system of distribution for moderate distances. An "accumulator" receives the water from the pumps and equalizes the pressure and flow. The incompressibility and fluidity of the liquid especially fit it for such use where the line of transmission is tortuous and irregular. Water taken from the street mains is often used, under pressures of from 30 to sometimes 100 pounds per square inch. The higher the safe pressure, the better, however; and a special supply at pressures exceeding 300 or even 700 and 1000 pounds is frequently found best. This system is often adopted for riveting-machines and other portable hydraulic tools and machines. Water-pressure is especially limited to cases in which the power is needed irregularly or at long intervals, when the work consists in the operation of reciprocating motors or machines; when irregular accumulation, as by wind-mills is practicable, and where exceptionably great pressures are demanded.

Compressed Air is used in cases somewhat similar to the preceding, and has been found especially suitable for mining operations, where water would be liable to freeze, and more particularly when high-speed rotation motors, and machines like rock-drills, are to be driven. It is, however, very wasteful of power, and in large operations it has often been found that no more than 25 per cent efficiency in transmission could be attained, high pressures being employed. This loss is in compression, largely; the loss by friction in pipes is not so large,

and should not exceed 5 per cent per mile. For driving small motors, however, the advantages of air are often great, and it is sometimes extensively used for this purpose. Its value in underground work is often greatest as supplying ventilation in otherwise inaccessible places. Moderate pressures, rarely exceeding 100 pounds, are used. Any form of motor that can be driven by steam may be used with air. For other than power purposes low pressures should be employed.

Electric transmission is finding extensive use in power-distribution, as is perhaps best illustrated by the systems of electric street-railway, and of supplying power for the minor industries. As in lighting, either the continuous current or the alternating may be used, although the former is the more generally adopted; and the continuous current may be furnished either by a generator direct or by a storage-battery.

A net efficiency of generator and motor, together not less than 75 per cent, should be attained. The loss on the line is very uncertain and variable. A tension of about 500 or 600 volts is usual.

The choice of this system is determined by a comparison of total costs. The losses enormously increase as the size of conductor in proportion to power transmitted diminishes. High tensions give great economy in this respect, while increasing leakage. Costs of conductors constitute a heavy tax on the system for long-distance transmission, and 2 to 5 miles may be taken as the ordinary range of practically attainable maxima.

This method has peculiar advantages for street-railway work, and has come into use mainly in that direction. Where lighting "plants" are already installed, it is often found that the addition of a power-system is economical and convenient, as well as otherwise desirable.

The cost of generating power from a fall of 80 feet as reported by Mr. Holt as obtained by dividing the cost of labor and lubricants (interest and depreciation are not included) by the horse-power demanded, amounts at present to

less than two-thirds of one cent per horse-power per hour, up to 100 horse-power.*

The advantages of electrical power for mining operations are: †

(1) It can be transmitted over long distances with small loss, making it possible to use power at such a distance from its source as would render it otherwise unavailable.

(2) The conductors for conveying electrical power require no appreciable space, are easily put in place and repaired, are easily tapped for branch circuits, and form a flexible system throughout.

(3) The electrical system is ideal in its cleanliness.

(4) The stations can be made to occupy a minimum of space.

A good illustration of the flexibility of the system is the diamond drill; in its use the conductors are unwound and strung up as the drill moves along, or taken down and coiled up as may be found convenient.

The waste of power from friction-resistance in the case of transmission by shafting is usually roughly stated at one per cent per one hundred feet; giving an ultimate limit at about ten thousand feet, or less than two miles, beyond which it is absolutely useless, and making it usually practically undesirable in lengths exceeding a few hundred feet. With electrical distribution this increase of waste with lengthening traverse is much less, and Beringer makes the total cost vary as the third or fourth root of the distance, increasing the more slowly as the power is the greater. Wire-rope transmission has a limit at three or four times that of shafting, or, commonly, five or six miles. Below a maximum of two to three miles this is, at present, the cheapest transmission; for higher figures the electrical is best. Practically the latter would, in most cases, be used beyond a mile.

Beringer has compared the four principal systems of power-

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* Trans. Am. Inst. Min. Engrs, Oct., 1891.

† Ibid.

transmission, by water, air, rope, and electricity, and finds the latter usually best.* Wire rope is found most economical for short distances, as between 100 feet and a half or three quarters of a mile, under the conditions assumed; but electricity is preferable beyond that maximum. Hydraulic and pneumatic systems cost much more, although the latter approximates the best figures at high powers and long distances, and all are more nearly alike as power transmitted and distances increase. Where air is wanted for ventilation, as in some mining operations, it often displaces all other methods. Electricity is now finding many applications in mining as well as in other power-transmissions.

It seems probable that these comparisons made with old and familiar systems may be altered somewhat, if not to an important extent, in favor of compressed-air transmission by adopting improved apparatus and methods—for example, as illustrated by the Popp distribution in Paris. By more effective spray-cooling at the point of compression, and by the adoption of a good compound type of compressor, Professor Riedler found it practicable to reduce the wastes from 43 to 12 per cent. By reheating the air at entrance into the engines at the other end of the system, Popp obtains, according to Professors Gutermuth and Weyrauch, a transformation of 70 per cent of the heat thus added into useful work, and a net gain of final efficiency of 30 per cent by raising the temperature of the working charge to 250° C. (482° F.). By compounding the motors and reheating between the two in series, Riedler reduced the consumption of air from 812 cubic feet per hour per brake horse-power to 646 with the steam-engine form and with rotary motors from 1059 to 847; the machines being in both cases of rude construction, inefficient type, and very small size. The investigators of this system consider it possible that it may prove, on the whole, the most economical method of power-transmission.†

* Kritische Vergleichung der Electrischen Kraft-übertragung, 1883.

† London Engineering, 1891; Scientific American Supplement, May 23, 1891.

The losses are always considerable. Thus, at St. Fargeau, Paris, air is compressed by the Popp Company to 6 atmospheres, sent 5 kilometres, and operating compressed-air engines at a pressure of 4½ atmospheres and an efficiency of 0.26. Compound compressors, however, may sometimes save a large proportion of the waste heat of compression, raising the efficiency of the compressor from 50 or 60, or from at most 70 to 85 or 90 per cent.* Good compressors and good engines should give at least 0.85 for the efficiency of the machine; and compressors should return from 70 to 90 per cent of the work expended upon them. Professor Unwin has computed a transmission for 10,000 H. P. twenty miles through mains 30 to 50 inches diameter, at an initial pressure of 75 to 190 pounds, and velocities of flow of 20 to 50 feet per second, with resultant efficiencies varying from 40 to 50 per cent; or if the air be heated at the engine, of 60 to 73 per cent.†

The costs and profits per mile run of street-railway power-transmission in Birmingham, G. B., all under a common management, were reported in 1891 as below: ‡

	Costs.	Earnings.	Profits.
Steam-plant	21.98	31.34	9.36
Wire cable	12.66	24.06	11.40
Electric	19.80	30.30	10.50
Horse	19.58	22.04	2.46

Running expenses only are included. Interest and repairs should be added. The order given is that of amount of traffic, the steam "cars" doing most work.

It will be usually found that each system is well adapted to a special set of economical conditions, and that neither can satisfactorily replace the other.

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* Riedler: Neue Erfahrungen über die Kraft-versorgung von Paris durch Druckluft; Berlin, 1891.

† Trans. Brit. Inst. C. E., 1891; No. 2548; vol. cv., Part III. See also vol. XCIII., p. 421.

‡ Engineering, Aug. 1891; Iron Age, Sept. 3. 1891.

Relay Power is demanded at times as accessory to the regular and usual motive power, either where, as with streams supplying water-power in varying quantity throughout the year, or where the load is itself varying and irregularly applied from day to day or season to season. In such instances steam-power is resorted to at times to supplement the temporary deficiency; and the kind, size, and economical value of the "relay" motor must be carefully considered.

In general, it may be said that if required for a longer time it must usually be more economical than if worked only a short time or for a small portion of the year, as is evident from the considerations studied in connection with problems of commercial efficiency. If used but seldom or for but a brief period, low first cost is the primary consideration; if for long periods, economy of operation must determine the size and character of the engine and its boilers.

Whatever the proportion of time in use, however, the engine should as far as possible conform to the primary requirement:

Minimum total cost of annual operation, including all the items enumerated when considering the problem of maximum commercial efficiency.

To this the following are accessory or subsidiary:

(1) Minimum first cost, consistent with the demanded efficiency.

(2) Efficiency adjusted to meet the primary demand above.

(3) Permanence of efficiency, despite the specially adverse circumstances of its use.

(4) Permanent good condition, though out of use.

(5) Stability of foundations and machinery of transmission.

(6) Minimum trouble and expense in "laying up" and again starting.

(7) Independence of skilled attendance.

The cost of machinery of transmission, whether by belting or gearing, and especially the loss of the power absorbed by its friction, often makes it advisable to avoid its use as far as

practicable, and to use separate engines for widely separated machines. This is especially the case in mills and other establishments in which the transmitting machinery constitutes a heavy and continuous load, while the driven machinery is operated only at intervals, and even, as is often the case, at long intervals and for brief periods of time. A judicious distribution of motors in such instances will often effect an enormous annual saving. It must, however, always be considered that, other things equal, several small engines will demand more steam than a single engine of equal total power.

The irregular demand at electric-lighting stations illustrates a peculiar case of what in a sense may also be termed "relay power," and the matter of subdivision of motive power and the problem arising out of it must often be settled by a study of existing "plants," and by reference to earlier experience. The experiments of Dr. Louis Bell being compared with those directly reported to the Author, indicated a total efficiency of but 25 per cent with large engines and of 37 per cent with small engines directly connected, the work being that of street-railways; but enormous variations are produced by differences in design, construction, and method of operation.*

In the Storage of Energy the method to be adopted must often be determined by the intensity and endurance demanded. The explosives give maximum intensity; the more common systems of engineering afford longest endurance. A rocket torpedo of 24 inches diameter is stated to develop 3000 H.P. for about 20 seconds; those of 15 and 12 inches, about 1400 and 750 H. P., respectively. The Whitehead torpedo is driven by compressed air, and develops about 50 H.P. for some minutes; the former attains 40 to 50 miles an hour as a maximum, the latter about 30, the range being about 1000 yards, the driving engines attaining a speed of about 1000 revolutions per minute.

Storage in fly-wheels is sometimes found practicable for high intensity and short period. The Howell torpedo employs

* Electrical World, Aug. 16, 1890; p. 103.

such a system. An "8-foot" torpedo, carrying a heavy cylindrical fly of 10.8" diameter of orbit of centre of gyration, at 10,000 revolutions per minute, stores 165 foot-tons of energy. The weights employed is here 110 pounds. At 12,000 revolutions the stored energy attains the amount of 350 foot-tons, nearly, and the range exceeds 800 yards, the speed at starting being 24 knots an hour.

The production of light demands in available form, according to Dewar, the storage or continued production of energy sufficient to sustain for the time required the following amounts of energy per candle-power:

Source.	Watts.	H. P.
Tallow demands	124	0.16
Wax "	94	0.12
Sperm "	86	0.11
Mineral oil	80	0.10
Vegetable oil	57	0.08
Coal-gas	68	0.09
Cannel-gas	48	0.06
Electricity (incandescent)	3	0.004
" (arc)	0.5	0.0007

APPENDIX.

I.

UNITED STATES STANDARD WEIGHTS AND MEASURES.

By T. C. MENDENHALL, Superintendent, U. S. Coast Survey.

LINEAR.

	Inches to millimetres.	Feet to Metres.	Yards to Metres.	Miles to Kilometres.
1 =	25.4000	0.304801	0.914402	1.60935
2 =	50.8001	0.609601	1.828804	3.21869
3 =	76.2001	0.914402	2.743205	4.82804
4 =	101.6002	1.219202	3.657607	6.43739
5 =	127.0002	1.524003	4.572009	8.04674
6 =	152.4003	1.828804	5.486411	9.65608
7 =	177.8003	2.133604	6.400813	11.26543
8 =	203.2004	2.438405	7.315215	12.87478
9 =	228.6004	2.743205	8.229616	14.48412

SQUARE.

	Sq. Ins. to Sq. Centimetres.	Square Ft. to Square Decimetres.	Square Yards to Square Metres.	Acres to Hectares.
1 =	6.452	9.290	0.836	0.4047
2 =	12.903	18.581	1.672	0.8094
3 =	19.355	27.871	2.508	1.2141
4 =	25.807	37.161	3.344	1.6187
5 =	32.258	46.452	4.181	2.0234
6 =	38.710	55.742	5.017	2.4281
7 =	45.161	65.032	5.853	2.8328
8 =	51.613	74.323	6.689	3.2375
9 =	58.065	83.613	7.525	3.6422

CUBIC.

	Cu. Ins. to Cubic Centimetres.	Cubic Feet to Cubic Metres.	Cubic Yards to Cubic Metres.	Bushels to Hectolitres.
1 =	16.387	0.02832	0.765	0.35242
2 =	32.774	0.05663	1.529	0.70485
3 =	49.161	0.08495	2.294	1.05727
4 =	65.549	0.11327	3.058	1.40969
5 =	81.936	0.14158	3.823	1.76211
6 =	98.323	0.16990	4.587	2.11454
7 =	114.710	0.19822	5.352	2.46696
8 =	131.097	0.22654	6.116	2.81938
9 =	147.484	0.25485	6.881	3.17181

CAPACITY.

	Fluid Drams to Millilitres or Cu. Centimetres.	Fluid Ounces to Millilitres.	Quarts to Litres.	Gallons to Litres.
1 =	3.70	29.57	0.94636	3.78544
2 =	7.39	59.15	1.89272	7.57088
3 =	11.09	88.72	2.83908	11.35632
4 =	14.79	118.30	3.78544	15.14176
5 =	18.48	147.87	4.73180	18.92720
6 =	22.18	177.44	5.67816	22.71264
7 =	25.88	207.02	6.62452	26.49808
8 =	29.57	236.59	7.57088	30.28352
9 =	33.28	266.16	8.51724	34.06896

WEIGHT.

	Grains to Milligrammes.	Avoirdupois Ounces to Grammes.	Avoirdupois Pounds to Kilogrammes.	Troy Ounces to Grammes.
1 =	64.7989	28.3495	0.45359	31.10348
2 =	129.5978	56.6991	0.90719	62.20696
3 =	194.3968	85.0486	1.36078	93.31044
4 =	259.1957	113.3981	1.81437	124.41392
5 =	323.9946	141.7476	2.26796	155.51740
6 =	388.7935	170.0972	2.72156	186.62089
7 =	453.5924	198.4467	3.17515	217.72437
8 =	518.3914	226.7962	3.62874	248.82785
9 =	583.1903	255.1457	4.08233	279.93133

1 chain	= 20.1169	metres
1 square mile	= 259	hectares
1 fathom	= 1.829	metres
1 nautical mile	= 1853.27	metres
1 foot = 0.304801 metre,	9.4840158	log
1 avoir. pound	= 453.5924277	gram.
15432.35639 grains =	1	kilogramme

The only authorized material standard of customary length is the Troughton scale belonging to the Coast Survey office, whose length at 59°.62 Fahr. conforms to the British standard. The yard in use in the United States is therefore equal to the British yard.

The only authorized material standard of customary weight is the Troy pound of the Mint. It is of brass of unknown density, and therefore not suitable for a standard of mass. It was derived from the British standard Troy pound of 1758 by direct comparison. The British Avoirdupois pound was also derived from the latter, and contains 7000 grains Troy.

The grain Troy is therefore the same as the grain Avoirdupois, and the pound Avoirdupois in use in the United States is equal to the British pound Avoirdupois. The British gallon = 4.54346 litres. The British bushel = 36.3477 litres.

	LINEAR.					SQUARE.					CUBIC.			
	Metres to Inches	Metres to Feet.	Metres to Yards.	Kilometres to Miles.		Sq. Centimetres to Sq. Inches.	Square Metres to Square Feet.	Square Metres to Square Yards.	Hectares to Acres.		Cu. Centimetres to Cu. Inches.	Cubic Decimetres to Cubic Inches.	Cubic Metres to Cubic Feet.	Cubic Metres to Cubic Yards.
1 =	39.3700	3.28083	1.093611	0.62137	1 =	0.1550	10.764	1.196	2.471	1 =	0.0610	61.023	35.314	1.308
2 =	78.7400	6.56167	2.187222	1.24274	2 =	0.3100	21.528	2.392	4.942	2 =	0.1220	122.047	70.629	2.616
3 =	118.1100	9.84250	3.280833	1.86411	3 =	0.4650	32.292	3.588	7.413	3 =	0.1831	183.070	105.943	3.924
4 =	157.4800	13.12333	4.374444	2.48548	4 =	0.6200	43.055	4.784	9.884	4 =	0.2441	244.093	141.258	5.232
5 =	196.8500	16.40417	5.468056	3.10685	5 =	0.7750	53.819	5.980	12.355	5 =	0.3051	305.117	176.572	6.540
6 =	236.2200	19.68500	6.561667	3.72822	6 =	0.9300	64.583	7.176	14.826	6 =	0.3661	366.140	211.887	7.848
7 =	275.5900	22.96583	7.655278	4.34959	7 =	1.0850	75.347	8.372	17.297	7 =	0.4272	427.163	247.201	9.156
8 =	314.9600	26.24667	8.748889	4.97096	8 =	1.2400	86.111	9.568	19.768	8 =	0.4882	488.187	282.516	10.464
9 =	354.3300	29.52750	9.842500	5.59233	9 =	1.3950	96.874	10.764	22.239	9 =	0.5492	549.210	317.830	11.771

	CAPACITY.						WEIGHT.							
	Millilitres or Cubic Centilitres to Fluid Drams.	Centilitres to Fluid Ounces.	Litres to Quarts.	Dekalitres to Gallons.	Hektolitres to Bushels.		Milligrammes to Grains.	Kilogrammes to Grains.	Hectogrammes (100 grammes) to Ounces Av.	Kilogrammes to Pounds Av.		Quintals to Pounds Av.	Milliers or Tonnes to Pounds Av.	Grammes to Ounces Troy.
1 =	0.27	0.338	1.0567	2.6417	2.8375	1 =	0.01543	15432.36	3.5274	2.20462	1 =	220.46	2204.6	0.03215
2 =	0.54	0.676	2.1134	5.2834	5.6750	2 =	0.03086	30864.71	7.0548	4.40924	2 =	440.92	4409.2	0.06430
3 =	0.81	1.014	3.1700	7.9251	8.5125	3 =	0.04630	46297.07	10.5822	6.61386	3 =	661.38	6613.8	0.09645
4 =	1.08	1.352	4.2267	10.5668	11.3500	4 =	0.06173	61729.43	14.1096	8.81849	4 =	881.84	8818.4	0.12860
5 =	1.35	1.691	5.2834	13.2085	14.1875	5 =	0.07716	77161.78	17.6370	11.02311	5 =	1102.30	11023.0	0.16075
6 =	1.62	2.029	6.3401	15.8502	17.0250	6 =	0.09259	92594.14	21.1644	13.22773	6 =	1322.76	13227.6	0.19290
7 =	1.89	2.368	7.3968	18.4919	19.8625	7 =	0.10803	108026.49	24.6918	15.43235	7 =	1543.22	15432.2	0.22505
8 =	2.16	2.706	8.4534	21.1336	22.7000	8 =	0.12346	123458.85	28.2192	17.63697	8 =	1763.68	17636.8	0.25721
9 =	2.43	3.043	9.5101	23.7753	25.5375	9 =	0.13889	138891.21	31.7466	19.84159	9 =	1984.14	19841.4	0.28936

By the concurrent action of the principal governments of the world, an International Bureau of Weights and Measures has been established near Paris. Under the direction of the International Committee, two ingots were cast of pure platinum-iridium in the proportion of 9 parts of the former to 1 of the latter metal. From one of these a certain number of kilogrammes were prepared, from the other a definite number of metre bars. These standards of weight and length were intercompared, without preference, and certain ones were selected as International prototype standards. The others were distributed by lot to the different governments, and are called national prototype standards. Those apportioned to the United States are in the keeping of this office.

The metric system was legalized in the United States in 1866.

The International Standard Metre is derived from the Metre des Archives, and its length is defined by the distance between two lines at 0° Centigrade, on a platinum-iridium bar deposited at the International Bureau of Weights and Measures.

The International Standard Kilogramme is a mass of platinum-iridium deposited at the same place, and its weight in vacuo is the same as that of the Kilogramme des Archives.

The litre is equal to a cubic decimetre of water, and it is measured by the quantity of distilled water which, at its maximum density, will counterpoise the standard kilogramme in a vacuum, the volume of such a quantity of water being, as nearly as has been ascertained, equal to a cubic decimetre.

II.

PROPERTIES OF THE ALLOYS OF COPPER AND ZINC.

*Comparison of several Authorities.**

Atomic Formula.	Composition of Original Mixture.		Composition by Analysis.		Specific Gravity.	Color.	Fracture.	Tenacity, Pounds per Square Inch.	Order of Ductility (Mallet).	Relative Ductility (Thurston).	Order of Malleability (Mallet).	Hardness (Mallet and Calvert and Johnson).	Order of Fusibility (Mallet).	Conductivity for Heat, silver = 100.	Conductivity for Electricity, silver = 100.	Authority.	Remarks.
	Cu.	Zn.	Cu.	Zn.													
	100	0	.....	.....	8.874	Red	Fibrous	27,800	....	30.8	....	.....	....	.....	.....	U. S. B.	
	100	0	.....	.....	8.667	Tile red	Earthy	55,104	8	.....	1	22	15	.....	.....	Ml.	
	100	0	.....	.....	8.921	.....	.....	.....	....	.....	....	.....	....	.....	93.16	Ma.	
	100	0	.....	.....	.....	.....	.....	.....	....	.....	....	30.1	....	81.1	.....	C. J.	
	100	0	.....	.....	8.952	.....	.....	.....	....	.....	....	.....	....	.....	.....	Mar.	Sheet copper.
	100	0	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	73.6	79.3	We.	
	100	0	.....	.....	8.672	.....	.....	24,252	....	.....	....	.....	....	.....	.....	We.	Mean of 9 samples.
	99.15	0.85	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	.....	.....	Bo.	Tombac for buttons.
	97.80	2.20	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	.....	.....	.....	Red Tombac for Vienna.
	97.50	2.50	97.83	1.88	8.791	Yellow red	Vesicular	27,240	....	118.9	....	.....	....	.....	.....	U. S. B.	
	97.20	2.50	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	.....	.....	Bo.	Railway axle boxes, porous.
	95	5	96.07	3.79	8.825	Yellow red	Vesicular	11,500	....	37.9	..	.....	..	.....	.....	U. S. B.	Defective bar.
	93.6	6.4	.....	.....	.....	.....	.....	.....	...	.....	..	.....	..	.....	.....	Bo.	Pinchback.
	92.5	7.5	92.32	7.68	8.746	Yellow red	Vesicular	.....	...	.....	....	.....	....	.....	.....	U. S. B.	
	92.5	7.5	.....	.....	.....	.....	.....	.....	...	.....	....	.....	....	.....	.....	Bo.	Bearings, Austria.
	92	8	.....	.....	.....	.....	.....	.....	...	.....	....	.....	....	.....	.....	Bo.	Red tombac of Paris.
	91	9	.....	.....	.....	.....	.....	.....	...	.....	....	.....	....	.....	.....	Bo.	Tombac.
$Cu_{10}Zn$..	90.72	9.28	.....	.....	8.605	Red yellow	Coarse crys.	27,104	6	.....	13	21	14	.....	.....	Ml.	
$Cu_{10}Zn$..	90.65	9.35	.....	9.60	8.834	.....	.....	.....	...	.....	....	.....	....	.....	.....	Rl.	Sp. gr. of ingot, 8.753.
	90	10.	90.56	9.42	8.773	Yellow red	Vesicular	.....	....	69.1	....	.....	....	.....	.....	U. S. B.	
	90	10.	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	.....	.....	Bo.	French oreide.
Cu_9Zn...	89.80	10.20	.....	.....	8.607	Red yellow	Finely crys.	25,760	4	.....	11	20	13	.....	.....	Ml.	
	88.89	11.11	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	27.3	25.5	We.	
Cu_8Zn...	88.60	11.40	.....	.....	8.633	Red yellow	Finely crys.	28,672	2	.....	10	19	12	.....	.....	Ml.	
	87.50	12.50	88.94	10.97	.....	Yellow red	Vesicular	.....	..	.....	....	.....	....	.....	.....	U. S. B.	
Cu_7Zn...	87.30	12.70	.....	.....	8.587	" "	Finely crys.	29,568	9	.....	9	18	11	.....	.....	Ml.	
	86.67	13.33	.....	.....	.....	.....	.....	.....	....	.....	....	.....	....	29.9	30.9	We.	

	86.38	13.61														Bo.	Very delicate castings.
	86	14														Bo.	Ornaments of Hanover.
	85.5	14.5														Bo.	French oreide.
Cu_6Zn...	85.4	14.6			8.591	Yellow red	Fine fibrous	31,584	5		8	17	10			Ml.	
Cu_6Zn..	85.34	14.66		14.90	8.710											Ri.	Sp. gr. of powder, 8.584.
	85.3	14.7														Bo.	Paris jewelry.
	85	15	89.80	10.06	8.656	Red yellow	Earthy			250.1						U. S. B.	
	85	15														Bo.	Tombac of Oker.
	84.6	15.4														Bo.	Gold leaf.
	84.21	15.79														Bo.	Tombac for buttons.
	83.99	17.1														Bo.	Bronze powder.
Cu_5Zn...	83.02	16.98			8.415	Red yellow	Finely crys.	30,688	11		2	16	9			Ml.	Bath metal.
Cu_5Zn...	82.95	17.05			8.673							427.08		71.5		C. J.	
	82.54	17.46														Bo.	Alloy of jewelry.
	82.5	17.5	82.93	16.98	8.633	Red yellow	Earthy	32,600		105.3						U. S. B.	
	82.46	17.54												31.1	29.2	We.	
	82.3	17.7														Bo.	Ornaments.
	80	20	181.91	17.90	8.598	Yellow	Earthy	32,670		113.1						U. S. B.	
Cu_4Zn...	79.65	20.35				Yellow red	Finely crys.	52,928	7		3	15	8			Ml.	Dutch brass.
Cu_4Zn...	79.56	20.44			8.650							468.75		66.6		C. J.	
Cu_4Zn...	79.51	20.49		21.80	8.638											Ri.	Sp. gr. of powder, 8.367.
	77.9	22.1														Bo.	Vienna gold-leaf.
	77.5	22.5	77.39	22.45	8.574	Yellow	Earthy	35,630		97.5						U. S. B.	
	75.7	24.3														Bo.	Bristol metal.
	75	25	76.65	23.08	8.528	Yellow	Earthy	30,520		76.7						U. S. B.	
	74.62	25.38														Bo.	Ship-sheathing.
Cu_3Zn..	74.58	25.42			8.307	Pale yellow	Finely crys.	29,344	10		4	14	7			Ml.	Rolled sheet brass.
Cu_3Zn...	74.48	25.52			8.576							468.75		63.8		C. J.	Brass of { 32 copper. 12 zinc.
	72.73	27.27														Bo.	
	72.5	27.5	73.20	26.47	8.465	Yellow	Earthy	31,580		88.7						U. S. B.	
	72	28														Bo.	Chrysorin.
	71.5	28.5														Bo.	Tombac.
	70.1	29.9														Bo.	Suitable for forg'g or leaf.
	70	30	71.20	28.54	8.444	Yellow	Earthy	30,510		77.8						U. S. B.	
	67.74	32.26												25.8	25.4	We.	
	67.54	32.46														Bo.	Suitable for forging.
	67.5	32.5	69.74	30.06	8.384	Yellow	Earthy	28,120		49.1						U. S. B.	
	67.2	32.8														Bo.	Bristol metal.
	66.7	33.3														Bo.	Chrysorin.
	66.6	33.4														Bo.	Common brass.
Cu_2Zn...	66.18	33.82			8.299	Full yellow	Finely crys	28,000	3		6	13	6			Ml.	
Cu_2Zn...	66.16	33.84			8.392											Cr.	
Cu_2Zn...	66.06	33.84			8.488							472.92		62.1		C. J.	

* Report of U. S. Board on Iron, Steel, and Other Metals; edited by R. H. Thurston, Washington, Gov't Print, vol. II., 1881. These tables were compiled for the Author by Mr. Wm. Kent.

PROPERTIES OF THE ALLOYS OF COPPER AND ZINC—*Continued.*

Atomic Formula.	Composition of Original Mixture. Cu.	Composition of Original Mixture. Zn.	Composition by Analysis. Cu.	Composition by Analysis. Zn.	Specific Gravity.	Color.	Fracture.	Tenacity, Pounds per Square Inch.	Order of Ductility (Mallet).	Relative Ductility (Thurston).	Order of Malleability (Mallet).	Hardness (Mallet, and Calvert and Johnson).	Order of Fusibility (Mallet).	Conductivity for Heat, Silver=100.	Conductivity for Electricity, Silver = 100.	Authority.	Remarks.
	66	34														Bo.	Suitable for forging.
	65.98	34.02		34.50	8.410											Ri.	Sp. gr. of powder, 8.390.
	65.4	34.6														Bo.	Good brass wire.
	65.3	34.7														Bo.	Mosaic gold.
	65	35	66.27	33.50	8.371	Red yellow	Earthy	37,800		72.8						U. S. B.	
	65	34.76														Bo.	Suitable for forging.
	63.5	36.5														Bo.	" " "
	62.5	37.5	63.44	36.36	8.411	Red yellow	Earthy	48,300		60.6						U. S. B.	
	61.25	38.75														Bo.	Strong solder for brass.
	60.8	39.2														Bo.	Bristol metal.
	60.16	39.71														Bo.	Suitable for forging.
	60	40	60.94	38.65	8.405	Red yellow	Earthy	41,065		49.0						U. S. B.	
	60	40															Muntz metal.
	59.5	40.5														Bo.	Ship-sheathing.
Cu_3Zn_2	59.36	40.64			8.224											Cr.	
Cu_3Zn_2	59.26	40.74		40.10	8.412											Ri.	Sp. gr. of powder, 8.329.
	58.33	41.77														Bo.	Suitable for forging.
	57.5	42.5	58.49	41.10	8.363	Red yellow	Earthy	50,450		12.1						U. S. B.	
	55	45														Bo.	Bath metal.
	55	45	55.15	44.44	8.283	Red yellow	Earthy	44,280		19.5						U. S. B.	
	54.9	45.1														Bo.	V'ry ductile brass (Storer).
	54	46														Bo.	" " " "
	52.5	47.5	54.86	44.78	8.301	Red yellow	Coarsely gran.	46,400		7.4						U. S. B.	
	50	50	49.66	50.14	8.291	" "	" "	30,990		3.1						U. S. B.	
CuZn	49.47	50.53			8.230	Full yellow	Coarse cryst.	20,608	12		5	12	6			Ml.	German brass.
CuZn	49.32	50.68			7.808							604.17		68.8		C. J.	
CuZn	49.23	50.76		50.30	8.304											Ri.	Sp. gr. of ingot, 8.263.
	47.5	52.5	48.95	50.82	8.216	Pink'h gray	Coarsely gran.	26,050		0.36						U. S. B.	
	45	55	47.56	52.28		" "	" "	24,150		0.26						U. S. B.	
	43	57														Bo.	Escutcheons of locks.
	42.5	57.5	43.36	56.22	8.034	Pink'h gray	Finely gran.	9,170		0.02						U. S. B.	
	40	60	41.30	58.12	8.061	Silver white	Vitr. conch.	3,727		0.01						U. S. B.	
Cu_2Zn_3	39.27	60.73		60.35	8.171											Ri.	Sp. gr. of ingot, 8.039.
	37.5	62.5	38.36	61.05	7.982	Silver white	Vitr. conch.	3,087		0.02						U. S. B.	
Cu_3Zn_5	36.88	63.12			7.939											Cr.	
	35	65	36.62	62.78	7.974	Silver white	Vitr. conch.	2,656		0.006						U. S. B.	

	33.34	66.66														Bo.	Strong solder for brass.
$CuZn_2$	32.85	67.15			8.283	De'p yellow	Coarse cryst.				7	10	6			Ml.	Watchmakers' brass.
$CuZn_2$	32.74	67.26			7.859							Broke		42.8		C. J.	
$CuZn_2$	32.66	67.34		64.80	8.048											Ri.	Sp. gr. of ingot, 7.796.
	32.5	67.5	35.68	63.71	7.966	Silver white	Vitr. conch.	2,397		0.11						U. S. B.	
Cu_8Zn_{17}	31.52	68.48			7.721	" "	Conchoidal	4,704	0		22	5	5			Ml.	Very brittle.
Cu_8Zn_{18}	30.30	69.70			7.836	" "	Vitr. conch.	4,928	0		23	6	5			Ml.	" "
	30	70	32.94	66.23	7.811	" "	" "	1,774		0.005						U. S. B.	
Cu_8Zn_{19}	29.17	70.83			8.019	Silver gray	Conchoidal	1,568	0		21	7	5			Ml.	Very brittle.
Cu_8Zn_{20}	28.12	71.88			7.603	Ash gray	Vitreous	7,168	0		19	3	5			Ml.	Brittle.
	27.5	72.5	29.20	70.17	7.766	Light gray	"	6,414		0.009						U. S. B.	
Cu_8Zn_{21}	27.10	72.90			8.058	Silver gray	Conchoidal	2,016	0		18	9	5			Ml.	Brittle.
Cu_8Zn_{22}	26.24	73.76			7.882	" "	"	1,792	0		20	8	5			Ml.	Very brittle.
	25.4	74.6														Bo.	Protect iron from rust.
Cu_8Zn_{23}	25.39	74.61			7.443	Ash gray	Finely cryst.	13,216	0		15	1	5			Ml.	Barely malleable.
	25	75			7.675	Bluish gray	Finely gran.	9,680		0.002						U. S. B.	
$CuZn_3$	24.64	75.36			7.736							Broke		53.1		C. J.	
$CuZn_3$	24.50	75.5			7.449	Ash gray	Finely cryst.	11,424	0		16	2	4			Ml.	Brittle.
	22.5	77.5	21.82	77.43	7.416	Bluish gray	Finely gran.	7,000		0.004						U. S. B.	
	20	80	20.81	77.63	7.418	" "	" "	9,000		0.002						U. S. B.	
	20	80														Bo.	Metal for buttons.
$CuZn_4$	19.65	80.35			7.371	Ash gray	Finely cryst.	4,256	0		14	4	3			Ml.	White button-metal.
$CuZn_4$	19.57	80.43			7.445							Broke		58.9		C. J.	
$CuZn_4$	19.52	80.48		79.30	7.863											Ri.	Sp. gr. of ingot, 7.215.
	17.5	82.5	17.49	81.62	7.225	Bluish gray	Finely gran.	5,350		0.003						U. S. B.	
$CuZn_5$	16.36	83.64			6.605	Very dark gray.	Finely cryst.	4,032	0		17	11	2			Ml.	Brittle.
$CuZn_5$	16.30	83.70			7.442							Broke		59.5		C. J.	
	15	85	14.19	85.10	7.163	Bluish gray	Finely gran.	8,500		0.004						U. S. B.	
	12.5	87.5	12.12	89.67	7.238	" "	" "	12,413		0.009						U. S. B.	
$CuZn_8$	10.82	89.18		89.90	7.315											Ri.	Sp. gr. of ingot, 7.299.
	10	90	10.30	88.88	7.253	Bluish gray	Finely gran.	14,450		0.10						U. S. B.	
	7.5	92.5	7.20	92.07	7.131	" "	" "	10,650		8.04						U. S. B.	
	5	95	4.35	94.59	7.108	" "	Finely cryst.	18,065		0.84						U. S. B.	
	2.5	97.5	2.45	96.43	7.080	" "	Tab. cryst.	11,400		2.67						U. S. B.	
	0	100			7.143	Blu'h white	" "	5,400		26.51						U. S. B.	
	0	100			6.895	Bluish gray	" "	34,048	13		12	23	1			Ml.	Brittle.
	0	100			7.148									29.02	29.39	Ma.	
	0	100												62.8		C. J.	
	0	100												28.1	27.3	We.	

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III.

PROPERTIES OF ALLOYS OF COPPER AND TIN.

Number.	Atomic Formula.	Composition of Original Mixture.		Composition by Analysis.		Specific Gravity.	Color.	Fracture.	Tenacity, Pounds per Square Inch.	Order of Ductility (Mallet).	Relative Ductility (Thurston).	Hardness (Mallet, and Calvert and Johnson).	Order of Malleability (Mallet).	Order of Fusibility (Mallet).	Conductivity for Heat. Silver = 100.	Conductivity for Electricity, Silver = 100.	Authority.	Remarks.
		Cu.	Sn.	Cu.	Sn.													
1		100.00	0.00			{ 8.791*a* 8.874*b* }	Copper red.	Fibrous	27,800		30.8						U. S. B.	{ *a* Specific gravity of bar. *b* Specific gravity of turnings from ingot. }
2		100.00	0.00			8.667	Tile red.	Earthy	55,104	1		10	[illegible]	16			M.	
3		100.00	0.00			8.921										93 16	Ma.	
4		100.00	0.00									301			81.1		C. J.	
5		100.00	0.00			8.794											Cr.	
6		100.00	0.00			8.921											Mar.	Cast copper.
7		100.00	0.00			8.952											Mar.	Sheet copper.
8		100.00	0.00												73.6	79.3	We.	
9		100.00	0.00			8.672			24,252								Na.	Mean of 9 samples.
10		98.59	1.41													62.46	Ma.	
11	$SnCu_{96}$	98.10	1.90	97.89	1.90	8.564	Red.	Vesicular			100.1						U. S. B.	Defective bar.
12		98.04	1.96														B.	Can be forged like copper.
13		98.00	2.00														La.	Ramrods for guns.
14		97.50	2.50			8.511	Red.	Vesicular									U. S. B.	Defective bar.
15		96.97	3.03														W.	Resists action of hydrochloric acid.
16	$SnCu_{48}$	96.27	3.73	96.06	3.76	8.649	Reddish yel.	Vesicular	32,000		70.3						U. S. B.	
17		96.00	4.00			8.947											Ri.	Annealed and compressed.
18		95.00	5.00				Golden yel.										W.	Hard malleable.
19		94.10	5.90														B.	Pieces of machines.
20		94.00	6.00			8.939											Ri.	Specific gravity after repeated tempering.
21		93.98	6.02														Ma.	
22	$SnCu_{25}$	93.17	6.83			8.820						602.8					C. J.	
23	$SnCu_{24}$	92.80	7.20			8.694	Reddish yel.	Vesicular	28,540		21.9						U. S. B.	

24		92.50	7.50			8.684	Reddish yel.	Vesicular	27,900		43.2						U. S. B.	
25		92.00	8.00														B.	Bronze for medals.
26		91.75	8.25														W.	Shows liquation.
27		91.74	8.26														B.	English ordnance.
28		91.70	8.30														B.	Ordnance-metal.
29		91.66	8.33														B.	8-pounder guns.
30	$SnCu_{20}$	91.49	8.51			8.793						639.58					C. J.	
31		91.30	8.70														B.	Toothed wheels.
32		90.91	9.09						32,093								Mus.	Prussian ordnance.
33		90.90	9.10														B.	Ordnance-metal.
34		90.73	9.27														B.	French ordnance.
35		90.10	9.90														B.	
36		90.00	10.00	90.27	9.58	8.669	Grayish yel.	Earthy	26,860		18.0						U. S. B.	Compressed ordnance-bronze.
37		90.00	10.00			8.875											De.	After repeated compression.
38		90.00	10.00			8.935											Ri.	Railroad-car bearings.
39		90.00	10.00														B.	Ordnance-metal.
40		89.30	10.70														B.	Small bar cast in iron mould.
41		89.29	10.71			8.953			37,688								Wa.	Small bar cast in clay mould.
42		89.29	10.71			8.313			25,783								Wa.	Mean of 12 gun-heads.
43		89.29	10.71			8.353			26,011								Wa.	Ordnance-metal.
44		89.23	10.77														B.	
45	$SnCu_{18}$	89.00	11.00			8.840											Ri.	
46	$SnCu_{15}$	88.97	11.03			8.825						772.92					C. J.	Mean of 83 gun-heads.
47		88.89	11.11			8.523			29,655								Wa.	
48		88.39	11.61													12.10	Ma.	Gun-metal.
49		88.00	12.00														La.	
50		87.65	12.35													10.15	Ma.	
51		87.50	12.50			8.648	Grayish yel.	Earthy	31,100		7.3						U. S. B.	
52		86.80	12.40														W.	
53	$SnCu_{13}$	86.57	13.43	87.15	12.73	8.681a 8.943b	Mottled white and yellow	Fine vesic.	29,430		13.9						U. S. B.	a Specific gravity of bar. b Specific gravity of fine turnings.
54		86.21	13.79			8.870											W.	Densest of all alloys (?) (Riche).
55		86.00	14.00														W.	Axle-bearing Seraing locomotive.
56		85.71	14.29						44,071								Mus.	
57		85.09	14.91													8.82	Ma.	
58	$SnCu_{10}$	84.33	15.67			8.870											Ri.	
59		84.39	15.68			8.832						916.66					C. J.	
60		84.29	15.71			8.561	Reddish yel., 1	Fine cryst.	36,064	2		8	6	15			Ml.	

PROPERTIES OF ALLOYS OF COPPER AND TIN—*Continued.*

Number.	Atomic Formula.	Composition of Original Mixture.		Composition by Analysis.		Specific Gravity.	Color.	Fracture.	Tenacity, Pounds per Square Inch.	Order of Ductility (Mallet).	Relative Ductility (Thurston).	Harness (Mallet, and Calvert and Johnson).	Order of Malleability (Mallet).	Order of Fusibility (Mallet).	Conductivity for Heat Silver = 100.	Conductivity for Electricity, Silver = 100.	Authority.	Remarks.
		Cu.	Sn.	Cu.	Sn.													
61		84.00	16.00														B.	Axle-bearings.
62		83.30	16.70														B.	Jewellers' punches.
63	$SnCu_9$	82.81	17.19			8.462	Reddish yel., 2	Fine cryst.	34,048	3		5	7	14			Ml.	
64		82.50	17.50			8.792	Reddish gray	"	36,200		0.32						U. S. B.	Strongest of series.
65		82.00	18.00			8.927											R.	Annealed and tempered.
66	$SnCu_8$	81.15	18.85			8.86											Ri.	
67	$SnCu_8$	81.10	18.90			8.459	Yel'wish red, 2	Fine cryst.	39,648	4		4	10	13			Ml.	
68				80.43	19.57	8.953											Th.	Chinese gong.
69		81.00	20.00			8.7											B.	Bells of Reichenhall, 300 years old.
70		80.00	20.00						35,739								Mus.	
71		80.00	20.00			8.850											Ri.	Annealed and tempered.
72		80.00	20.00			8.955											Ri.	After repeated compression.
73		80.00	20.00	80.95	18.84	8.740	Reddish gray	Fine gran.	32,980		0.40						U. S. B.	
74		79.20	20.80			8.927											Ri.	Annealed and tempered.
75	$SnCu_7$	79.02	20.98			8.90											Ri.	
76	$SnCu_7$	78.97	20.03			8.728	Yel'wish red, 1	Vitr. conch.	30,464	5		3	11	12			Ml.	
77		78.00	20.00														B.	Best bell-metal.
78		77.50	22.50			8.917	Pinkish gray	Fine gran.	24,650		0.03						U. S. B.	
79	$SnCu_6$	76.32	23.68	76.64	23.24	8.565		Smooth	22,010		0.02						U. S. B.	
80	$SnCu_6$	76.31	23.69			8.91											Ri.	
81	$SnCu_6$	76.29	23.71			8.750	Bluish red	Vitreous	21,728	0		2	12	11			Ml.	
82		76.20	23.80			9.1(?)											B.	Church-bell in Reichenhall, 600 years old.
83		75.20	24.80														B.	Swiss clock-bells, brittle.
84		75.00	25.00			8.879	Reddish white										W.	
85	$SnCu_5$	42.91	27.09			8.87											Ri.	
86	$SnCu_5$	72.90	27.10			8.965						Broke			20.7		C. J.	

87	$SnCu_5$..	72.80	27.20			8.575	Bluish red	Conchoidal	10,976	0		1	13	10			Ml.	
88		72.50	27.50			8.925	Reddish white	"	6,493		0.003						U. S. B.	
89		70.00	30.00	69.84	29.89	8.932	White	"	5,585		0.008						U. S. B.	
90		68.82	31.18														B.	Mirror of Telescope.
91	$SnCu_4$..	68.28	31.72			8.80											Ri.	
92	$SnCu_4$..	68.27	31.73			8.948						Broke			15.5		C. J.	
93	$SnCu_4$..	68.25	31.75	68.58	31.26	8.938	White	Conchoidal	1,620								U. S. B.	
94	$SnCu_4$..	68.21	31.79			8.400	Ash gray	"	1,568			6	14	9			Ml.	
95		68.21	31.70							0							B.	Mirror Metal.
96		67.50	32.50			8.907	White	Conchoidal	2,536		0.009						U. S. B.	
97		66.67	33.33				Steel gray										W.	Hard, uniform.
98		66.67	33.33						1,017								Mus.	
99		65.00	35.00	65.34	34.47	8.947	Bluish gray		2,201		0.002						U. S. B.	
100		62.50	37.50			8.956	Dark gray	Rad. cryst.	1,561		0.001						U. S. B.	
101	$SnCu_3$..	61.79	38.21			8.954						Broke			49.4		C. J.	
102	$SnCu_3$..	61.79	38.21			8.96											Ri.	
103	$SnCu_3$..	61.71	38.29			8.970	Dark gray	Rough, stony	688		0.003						U. S. B.	Greatest density, weakest.
104	$SnCu_3$..	61.69	38.31			8.539	Dark gray	Tabul'r cryst.	1,120	0		7	16	8			Ml.	
105		60.00	40.00														B.	White bell-metal.
106		57.50	42.50			8.781	Light gray	Stony	1,377		0.001						U. S. B.	
107	Sn_5Cu_{12}	56.32	43.68	56.70	43.17	8.682	"	"	1,455		0.003						U. S B	
108		52.50	47.50			8.643	"	"			0.001						U. S. B.	Weakest under transverse stress.
109	$SnCu_2$..	51.84	48.16			8.57											Ri.	
110	$SnCu_2$					8.512											Cr.	
111	$SnCu_2$..	51.83	48.17			8.533						Broke			42.8		C. J.	
112	$SnCu_2$..	51.80	48.20	51.62	48.09	8.560	Light gray		2,555		0.003						U. S. B.	Remarkable for liquation.
113	$SnCu_2$.	51.75	48.25			8.416	Grayish white	Vit. conch.	3,808	0		9	15	7			Ml.	
114		50.00	50.00						725								Mus.	
115		50.00	50.00			8.79	Bluish white										W.	Very slightly malleable.
116	Sn_7Cu_2	47.95	52.05	47.61	52.14	8.442	Grayish white	Fine grain	1,525		0.003						U. S. B.	
117		47.50	52.50			8.446	"	Crystal	2,407		0.003						U. S. B.	
118	Sn_2Cu_3	44.67	55.33			8.30											Ri.	
119	Sn_2Cu_3	44.63	55.37	44.52	55.28	8.312	Grayish white	Crystal	3,010		0.003						U. S. B.	
120		42.50	57.50			8.437	"	"	2,098		0.006						U. S. B.	
121	Sn_3Cu_4	41.74	58.26	42.38	57.30	8.302	"	"	3,910		0.007						U. S. B.	
122		40.00	60.00														B.	White bell-metal.
123	Sn_5Cu_6	39.20	60.80	38.37	61.32	8.182	Grayish white	Crystal	2,820		0.005						U. S. B.	
124		37.50	62.50			8.101	"	"	2,400		0.004						U. S. B.	
125	$SnCu$...	34.99	65.01			8.12											Ri.	
126	$SnCu$...	34.98	65.02			7.992						Broke			41.5		C. J.	
127	$SnCu$...					8.072											Cr.	
128	$SnCu$...	34.95	65.05	34.22	65.80	8.013	Grayish white	Crystal	3,371		0.005						U. S. B.	
129	$SnCu$...	34.92	65.08			8.056	"	Tabul'r cryst.	3,136	0		11	9	6			Ml.	
130		33.33	66.67				White										W.	Brittle, uniform.
131		32.50	67.50			7.931	Grayish white	Crystal	2,322		0.014						U. S. B.	

PROPERTIES OF ALLOYS OF COPPER AND TIN—*Continued.*

Number.	Atomic Formula.	Composition of Original Mixture.		Composition by Analysis.		Specific Gravity.	Color.	Fracture.	Tenacity, Pounds per Square Inch.	Order of Ductility (Mallet).	Relative Ductility (Thurston).	Hardness (Mallet and Calvert and Johnson).	Order of Malleability (Mallet).	Order of Fusibility (Mallet).	Conductivity for Heat, Silver = 100.	Conductivity for Electricity, Silver = 100.	Authority.	Remarks.
		Cu.	Sn.	Cu.	Sn.													
132	Sn_4Cu_3	28.72	71.28	25.85	73.80	7.918	Grayish white	Crystal	1,648		0.011						T.	
133		27.50	72.50			7.915	"	"	4,380		0.018						T.	
134		25.00	75.00			7.813	Bluish white	"									W.	
135	Sn_5Cu_3	24.38	75.62	23.35	76.29	7.835	Grayish white	Fine cryst.	6,775		0.03						T.	
136		22.50	77.50			7.774	"	"	5,000		0.12						T.	
137		21.74	78.26	21.38	77.63	7.530	Whitish										W.	Yellow, greenish white, shining.
138	Sn_2Cu..	21.21	78.79			7.738						135.42			43.1		C. J.	
139	Sn_2Cu..	21.21	78.79			7.740											Ri.	
140	Sn_2Cu..	21.18	78.82	20.25	79.63	7.770	Grayish white	Crystal	4,337		0.06						T.	
141	Sn_2Cu..	21.15	78.85			7.387	"	Coarse cryst.	8,730	0		12.	8	5			Ml.	
142	Sn_5Cu_2	17.68	82.32			7.652											Cr.	
143		17.50	82.50			7.690	Grayish white	Crystal	2,816		0.20						T.	
144		16.40	83.60													12.76	Ma.	
145	Sn_3Cu..	15.21	84.79			7.530											Ri.	
146	Sn_3Cu..	15.21	84.79			7.606						104.17			42.3		C. J.	
147	Sn_3Cu..	15.19	84.81	15.08	84.62	7.657	Grayish white	Crystal	6,520		0.92						T.	
148	Sn_3Cu..	15.17	84.83			7.447	"	Coarse cryst.	6,944	0		13.	15	4			Ml.	
149		12.50	87.50			7.543	"	Crystal	3,798		4.71						T.	
150	Sn_4Cu..	11.86	88.14			7.558						95.81			40.6		C. J.	
151	Sn_4Cu..	11.84	88.16	11.49	88.47	7.552	Grayish white	Crystal	6,380		7.08						T.	
152	Sn_4Cu..	11.84	88.16			7.500											Ri.	
153	Sn_4Cu..	11.82	88.18			7.472	Grayish white	Coarse cryst.	6,944	8		14.	4	3			Ml.	
154	Sn_5Cu..	9.73	90.27			7.517						83.33			39.6		C. J.	
155	Sn_5Cu..	9.73	90.27			7.520											Ri.	
156	Sn_5Cu..	9.70	90.30	8.57	91.39	7.487	Grayish white	Granular	6,450		23.47						T.	
157	Sn_5Cu..	9.68	90.32			7.442	"	Earthy	3,360	6		15.	3	2			Ml.	
158		9.09	90.91			7.472											W.	Slightly malleable.
159		7.50	92.50			7.417	Grayish white	Granular	6,069		40.06						U. S. B.	

160	6.43	93.57													12.03	Ma.
161 $Sn_{12}Cu$	4.29	95.71	3.72	96.31	7.360	Grayish white	Granular	4,780		56.77						U. S. B.
162	2.50	97.50			7.342	"	Fibrous	5,600		121.9						U. S. B.
163 $Sn_{48}Cu$	1.11	98.89	0.74	99.02	7.305	"	"	3,650		133.9						U. S. B.
164 $Sn_{96}Cu$	0.56	99.44	0.32	99.46	7.299	"	"	4,475		208.8						U. S. B.
165 Sn.	0.	100.00			7.293	"	"	3,500		219.8						U. S. B.
166	0.	100.00			7.291	White	"	6,040	7		16.	1	1			Ml.
167	0.	100.00			7.297			2,122								Wa.
168	0.	100.00			7.294										11.45	Ma.
169	0.	100.00			7.305											Cr.
170	0.	100.00									27.			42.2		C. J.
171	0.	100.00												15.2	17.0	We.

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IV.

WEIGHT OF METALS OF SIMILAR FORM AND DIMENSIONS, IN KILOGRAMMES AND IN POUNDS.—Hoare.*

		Description of Metal.		Rolled Iron.	Forged Iron.	Cast Iron.	Bessemer Steel.	Rolled Steel.
		Specific Gravity and × by 1000, showing No. of Ounces in C Foot		7.680	7.782	7.271	7.872	7.833
Cubical and Rectangular	1	1 cubic foot	lbs.	480.	486.4	454.44	492.	489.56
	2	1 " "	cwt.	4.2857	4.343	4.058	4.393	4.371
	3	1 " inch	lbs.	.277	.2815	.263	.285	.2833
	4	Number of cubic inches in 1 lb.	No.	3.6	3.55	3.8	3.509	3.53
	5	1 square foot 1.000 inch thick	lbs.	40.	40.53	37.87	41.	40.797
	6	1 " " 0.125⅛ " "	lbs.	5.	5.067	4.73	5.125	5.098
	7	1 " " 0.001 1/1000 " "	lbs.	.04	.0405	.0379	.041	.0408
	8	1 " " 1 millimetre "	lbs.	1.5748	1.6	1.491	1.614	1.6
	9	1 cubic metre	k'gms.	7680.	7782.	7271.	7872.	7833.
	10	1 " "	lbs.	16931.328	17156.197	16029.62	17363.27	17277.248
	11	1 " centimetre	lbs.	.01693	.01716	.01603	.01736	.01728
	12	1 square metre 1 millimetre thick	k'gms.	7.680	7.782	7.271	7.872	7.833
	13	1 " " 1 " "	lbs.	16.931	17.156	16.0290	17.3633	17.27725
Cylindrical and Circular	1	1 cylindrical foot	lbs.	377.	382.018	356.92	386.42	384.5
	2	1 " "	cwt.	3.357	3.411	3.1868	3.45	3.433
	3	1 " inch	lbs.	.218	.2211	.2066	.2236	.2225
	4	Number of cylindrical inches in 1 lb.	No.	4.59	4.5	4.84	4.472	4.5
	5	1 circular foot 1.000 inch thick	lbs.	31.416	31.835	29.74	32.2	32.04
	6	1 " " 0.125 " "	lbs.	3.927	3.98	3.72	4.025	4.
	7	1 " " 0.001 " "	lbs.	.03142	.0318	.02974	.0322	.032
	8	1 " " 1 millimetre	lbs.	1.237	1.2533	1.171	1.268	1.26
	9	1 cylindrical metre	k'gms.	6031.872	6112.	5710.643	6182.669	6152.038
	10	1 " "	lbs.	13297.865	13474.5	12589.684	13629.21	13562.782
	11	1 " centimetre	lbs.	.01329	.01348	.01259	.03637	.01357
	12	1 circular metre 1 millimetre thick	k'gms.	6.0319	6.112	5.7106	6.1827	6.152
	13	1 " " 1 " "	lbs.	13.2978	13.4745	12.5897	13.6292	13.5628
Spherical	1	1 spherical foot	lbs.	250 56	254.68	237.94	257.82	256.33
	2	1 " inch	lbs.	.145	.1474	.1377	.1492	.1483
	3	Number of spherical inches in 1 lb.	No.	6.9	6.784	7.262	6.7	6.74
	4	1 spherical metre	k'gms.	4021.248	4074.65	3807.096	4121.78	4101.359
	5	1 " "	lbs.	8865.24	8982.97	8393.12	9086.876	9041.856
	6	1 " centimetre	lbs.	.00886	.00898	.00839	.00909	.00904
Bars 1-in. Cross-section	1	1-inch square bar 1 foot long	lbs.	3.33	3.37	3.156	3.416	3.4
	2	1 " round " 1 " "	lbs.	2.618	2.653	2.48	2.683	2.67
	3	1 " octagon " 1 " " (across the flat)	lbs.	2.7612	2.798	2.614	2.83	2.816
	4	1 " hexagon " 1 " "	lbs.	2.9	2.939	2.746	2.973	2.958
	5	1 " 3-square " 1 " " (on the edge)	lbs.	1.443	1.462	1.366	1 479	1.472

* Hoare's Iron and Steel; Carden, Lockwood & Co.

		Description of Metal.		Rolled Copper.	Rolled Brass (average).	Rolled Lead.	Water *(see note).		
		Specific Gravity and × by 1000, showing No. of Ounces in C Foot.		8.790	8.400	11.340	1.000		
Cubical and Rectangular	1	1 cubic foot	lbs.	549.375	525.	709.25	62.425	1	Cubical and Rectangular
	2	1 " "	cwt.	4.905	4.6884	6 333	.557	2	
	3	1 " inch	lbs.	.318	.3038	.4104	.03612	3	
	4	Number of cubic inches in 1 lb.	No.	3.144	3.291	2.43	27.685	4	
	5	1 square foot 1.000 inch thick	lbs.	45.76	43.73	58.75	5.208	5	
	6	1 " " 0.125 1/8 " "	lbs.	5.72	5.466	7.314	.651	6	
	7	1 " " 0 001 1/1000 " "	lbs.	.0458	.04373	.05875	.00521	7	
	8	1 " " 1 millimetre "	lbs.	1.8	1.722	2.313	.205	8	
	9	1 cubic metre	k'gms.	8790.	8400.	11340.	1000.	9	
	10	1 " "	lbs.	19378.434	18518.64	25000.16	2204.6	10	
	11	1 " centimetre	lbs.	.01938	.01852	.025	.0022	11	
	12	1 square metre 1 millimetre thick	k'gms.	8.79	8.4	11.34	1.	12	
	13	1 " " 1 " "	lbs.	19.3784	18.5186	25.	2.2046	13	
Cylindrical and Circular	1	1 cylindrical foot	lbs.	431.479	412.335	557.045	49.1	1	Cylindrical and Circular
	2	1 " "	cwt.	3.8525	3.681	4.08	.032	2	
	3	1 " inch	lbs.	.2497	.2386	.3223	.02842	3	
	4	Number of cylindrical inches in 1 lb.	No.	4.	4.191	3.103	35.15	4	
	5	1 circular foot 1.000 inch thick	lbs.	35.952	34.356	46.416	4.09	5	
	6	1 " " 0.125 " "	lbs.	4.494	4.294	5.802	.5101	6	
	7	1 " " 0.001 " "	lbs.	.0359	.0343	.0464	.00409	7	
	8	1 " " 1 millimetre	lbs.	1.415	1.353	1.827	.15079	8	
	9	1 cylindrical metre	k'gms.	6903.666	6597.36	8906.436	785.4	9	
	10	1 " "	lbs.	15219.82	14544.54	19635.13	1732.3568	10	
	11	1 " centimetre	lbs.	.0152	.01454	.01963	.00173	11	
	12	1 circular metre 1 millimetre thick	k'gms.	6.9037	6.59736	8.9064	.7854	12	
	13	1 " " 1 " "	lbs.	15.2198	14.5445	19.635	1.73236	13	
Spherical	1	1 spherical foot	lbs.	287.652	274.89	371.363	32.73	1	Spherical
	2	1 " inch	lbs.	.1665	.159	.2149	.0189	2	
	3	Number of spherical inches in 1 lb.	No.	6.	6.3	4.653	52.9	3	
	4	1 spherical metre	k'gms.	4602.444	4398.24	5937.624	523.6	4	
	5	1 " "	lbs	10146.548	9696.36	13090.07	1154.9	5	
	6	1 " centimetre	lbs.	.01015	.00969	.01309	.00115	6	
Bars 1-in. Cross-section	1	1-inch square bar 1 foot long	lbs.	3.8151	3.645	4.925	.434	1	Bars 1-in. Cross-section
	2	1 " round " 1 " "	lbs.	2.996	2.863	3.868	.34104	2	
	3	1 " octagon " 1 " " (across the flat)	lbs.	3.16	3.02	4.08	.3595	3	
	4	1 " hexagon " 1 " "	lbs.	3.32	3.172	4.286	.3777	4	
	5	1 " 3-square " 1 " " (on the edge)	lbs.	1.652	1.578	2.131	.1879	5	

V.

REPORT OF BOILER TEST.

Made by...

..........................*N. Y.*, ...189..

Kind of Boiler....................... ...*Made by*.......................

	Duration of Trial.	**Hours.**
Dimensions.	Grate-surf., length, width.....	Sq. ft.
	Water-heating surface.........	"
	Superheating surface..........	"
	Area for draught (calorim.)....	"
	Area, chimney................	"
	Height, chimney................	Ft.
	Ratio heating to grate-surface...	"
	Ratio air-space to grate-surface.....	
Pressure.	Barometer, inches mercury..........	
	Steam-gauge, pounds.	
	Draught-gauge, inches water.......	
	Absolute steam-pressure, pounds....	
Temperature.	External air............	Degrees F.
	Boiler-room........	"
	Flue...............	"
	Furnace................	"
	Feed-water....	"
	Steam..................	"
Fuel.	Total coal consumed........	Pounds.
	Moisture in coal...........	Per cent.
	Dry coal consumed........	Pounds.
	Total refuse, dry..........	"
	Total refuse, dry...	Per cent.
	Total combustible......	Pounds.
	Dry coal, per hour..........	"
	Combustible, per hour......	"
Fuel per Hour.	Dry coal per square foot of grate..	"
	Combustible per square foot of grate..	"
	Dry coal per square foot of grate..........	Heating-surface.
	Combustible per square foot of grate..........	Heating-surface.
	Quality of steam....	Per cent.
	Superheat................	Degrees.
Tot. Water.	Total wt. of water used.....	Pounds.
	(by meter)...	Cu. ft.
	Total evap., dry steam......	Pounds.
	Factor of evaporation	
	Total from and at 212°.....	Pounds.

Water per Hr.	Amount used....	Pounds.
	Evaporated, dry steam......	"
	Evap. from and at 212°......	"
Evaporation.	*Per Pound of Fuel.*	
	Actual.....	Pounds.
	Equiv. from and at 212°......	"
	Per Pound of Combustible.	
	Actual......................	Pounds.
	Equiv. from and at 212°.....	"
	Per Sq. Ft. Heating-surface per Hr.	
	Actual.	Pounds.
	Equiv. from and at 212°.....	"
	From 100° F. to 70 Pounds by Gauge.	
	Per Pound of Fuel....	
	Per Pound of Combustible..........	
	Per ½-pound of Fuel................	
Evaporation, Per Hour.	*Per Square Foot of Grate.*	
	Actual, from feed-water temperature....	Pounds.
	Equiv. from and at 212°....	"
	Per Sq. Ft. of Water-heating Surface.	
	Actual........	
	Equiv. from and at 212°.	
	Per Sq. Ft. of Least Draught-area.	
	Actual............................	
	Equiv. from and at 212°.............	
H. P.	On basis 34½ lbs. eq. evap. per hr.....	
	Builders' rating..........	
	Ratio of commercial to builders' rating.....	
	Heat generated per hour...	B. T. U.
	Heat absorbed per hour....	"
	Efficiency of boiler.	Per cent.
	Efficiency of furnace	"

NOTE.—Actual evaporation signifies the evaporation from feed-water temperature to dry steam at gauge-pressure. It is apparent evaporation corrected for calorimeter determinates

DESCRIPTION OF BOILER—REMARKS.

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VI.

COURSE OF INSTRUCTION IN MECHANICAL ENGINEERING.

[ROBERT H. THURSTON—July, 1871.]

I.

MATERIALS USED IN ENGINEERING.—Classification, Origin, and Preparation (where not given in course of Technical Chemistry), Uses, Cost.

Strength and Elasticity.—Theory (with experimental illustrations) reviewed, and tensile, transverse, and torsional resistance determined.

Forms of greatest strength determined. *Testing* materials.

Applications.—Foundations, Framing in wood and metal.

FRICTION.—Discussion from Rational Mechanics, reviewed and extended.

Lubricants treated with materials above.

Experimental determination of "coefficients of friction."

II.

TOOLS.—Forms for working wood and metals. Principles involved in their use.

Principles of pattern-making, moulding, smith and machinists' work, so far as they modify design.

Exercises in Workshops in mechanical manipulation.

Estimates of *cost* (stock and labor).

MACHINERY AND MILL-WORK.—Theory of machines. Construction. Kinematics applied. Stresses, calculated and traced. Work of machines. Selection of materials for the several parts. Determination of *proportions* of details, and of *forms* as modified by difficulties of construction.

Regulators, Dynamometers. Pneumatic and Hydraulic machinery. Determining *moduli* of machines.

POWER, transmission by gearing, belting, water, compressed air, etc.

LOADS, transportation.

III.

HISTORY AND PRESENT FORMS OF THE PRIME-MOVERS.

Windmills, their theory, construction, and application.

Water-wheels. Theory, construction, application, testing, and comparison of principal types.

Air, Gas, and Electric Engines, similarly treated.

STEAM-ENGINES.—Classification. [Marine (merchant) Engine assumed as representative type.] Theory. Construction, including general design, form and proportion of details.

Boilers similarly considered. Estimates of *cost*.

Comparison of principal types of Engines and Boilers.

Management and repairing. Testing and recording performance.

IV.

MOTORS APPLIED to Mills. Estimation of required power and of *cost*.

Railroads. Study of Railroad machinery.

Ships. Structure of Iron Ships and Rudiments of Naval Architecture and Ship Propulsion.

PLANNING Machine-shops, Boiler-shops, Foundries, and manufactories of textile fabrics. Estimating *cost*.

LECTURES BY EXPERTS.

GENERAL SUMMARY of principal facts, and natural laws, upon the thorough knowledge of which successful practice is based; and general *résumé* of principles of business which must be familiar to the practising engineer.

V.

GRADUATING THESES.

Graduation.

Accompanying the above are courses of instruction in higher mathematics, graphics, physics, chemistry, and the modern languages and literatures.

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